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Two Engineers

By Josh Weller

I KNOW a certain engineer; his hair is white as snow;
He's working in the same old plant he worked in years ago.
And when I try to tell him how his steam plant should be run
He comes right back and says, "I've been here forty years, my son."

He has a nasty laundry job, and helps to wash the duds;
He's always on the jump between the coal pile and the suds.
The world has moved, the times have changed, some things have had their day,
And yet he keeps on doing things in just the same old way.

"The water's bad," he doses it with Epsom salts and such,
Then wonders why his boiler has the belly-ache so much.
He burns two tons of coal each day when one would be enough,
And kicks on all the coal he gets from anthracite to "duff."

"I've been here forty years, my son."

And now he has the largest plant and biggest pay in town.

He swears by all that's good and bad and by the sun and moon,
And by the prophet's beard, by gosh, and by the great horn spoon,
That if he had a thousand boys, the one he held most dear
Of all that mighty host of sons should be an engineer.

I know another engineer, and this is what I know—
He started in a laundry plant just ten short years ago,
He boned away at books and things; they couldn't keep him down,
And now he has the largest plant and biggest pay in town.

It's one man's lot to buck the coal and wash the shirts and socks,
Another's lot to get the job and gather in the rocks.
Some men are paid for what they do and some for what they know,
And if you'll look around a bit, I think you'll find it so.

McIntosh & Seymour 500-Hp. Diesel Engine

By F. R. Low

SYNOPSIS—Description of 500-hp. vertical, 4-cylinder engine built by the McIntosh & Seymour Corporation under license from the Swedish Diesel Engine Co.

The idea is prevalent that the Diesel engine is a complicated piece of machinery, but when one looks at the reproduced photographs on the opposite page he sees all that there is of a Diesel-engine plant; whereas a steam engine must have a boiler, heater, feed pump, injector, condenser and an air pump. The man that stands upon the platform of a Diesel engine has the whole plant under his eye. Furthermore, the engine illustrated is really four engines, all alike and each extremely simple—a shaft, connecting-rod, piston, inlet valve, exhaust valve and a fuel valve, with mechanism for operating and controlling the valves, constitute the entire list of working parts for each one of these engines.

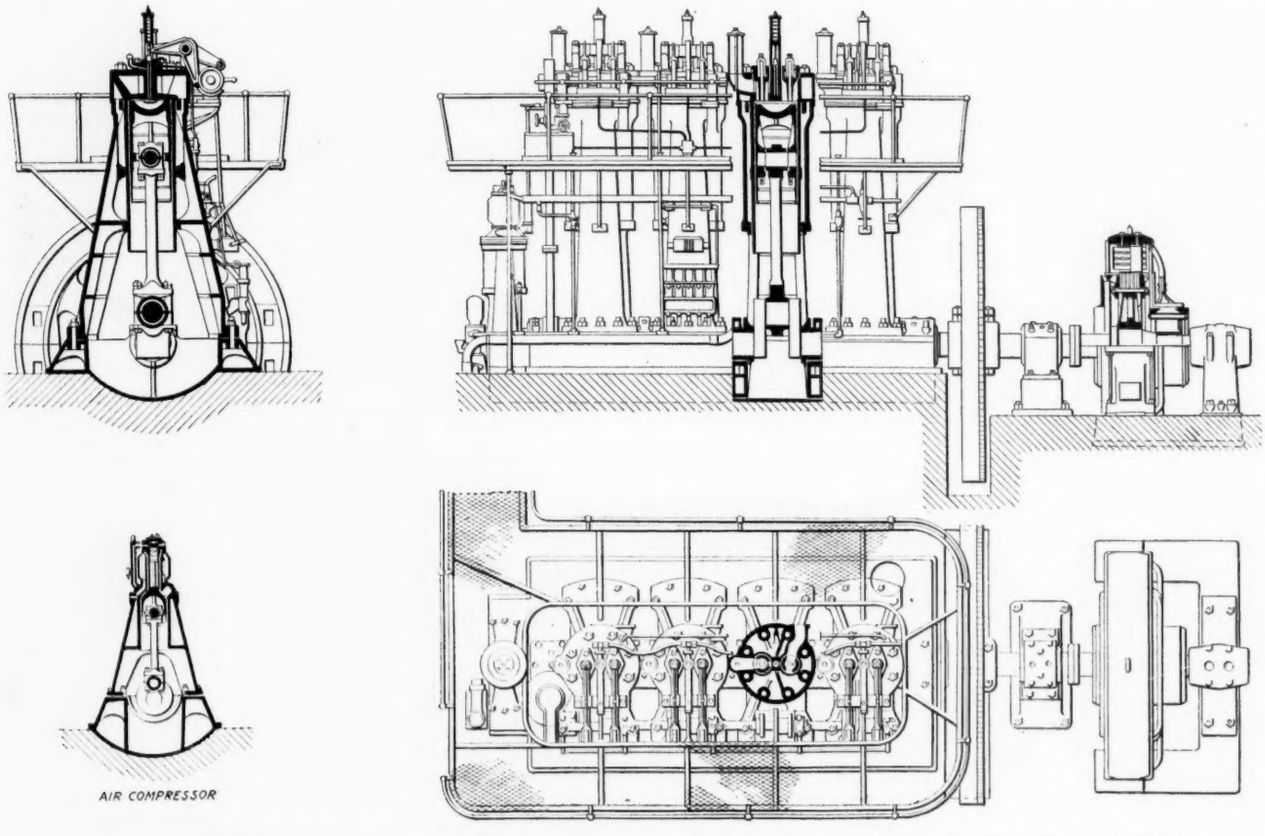
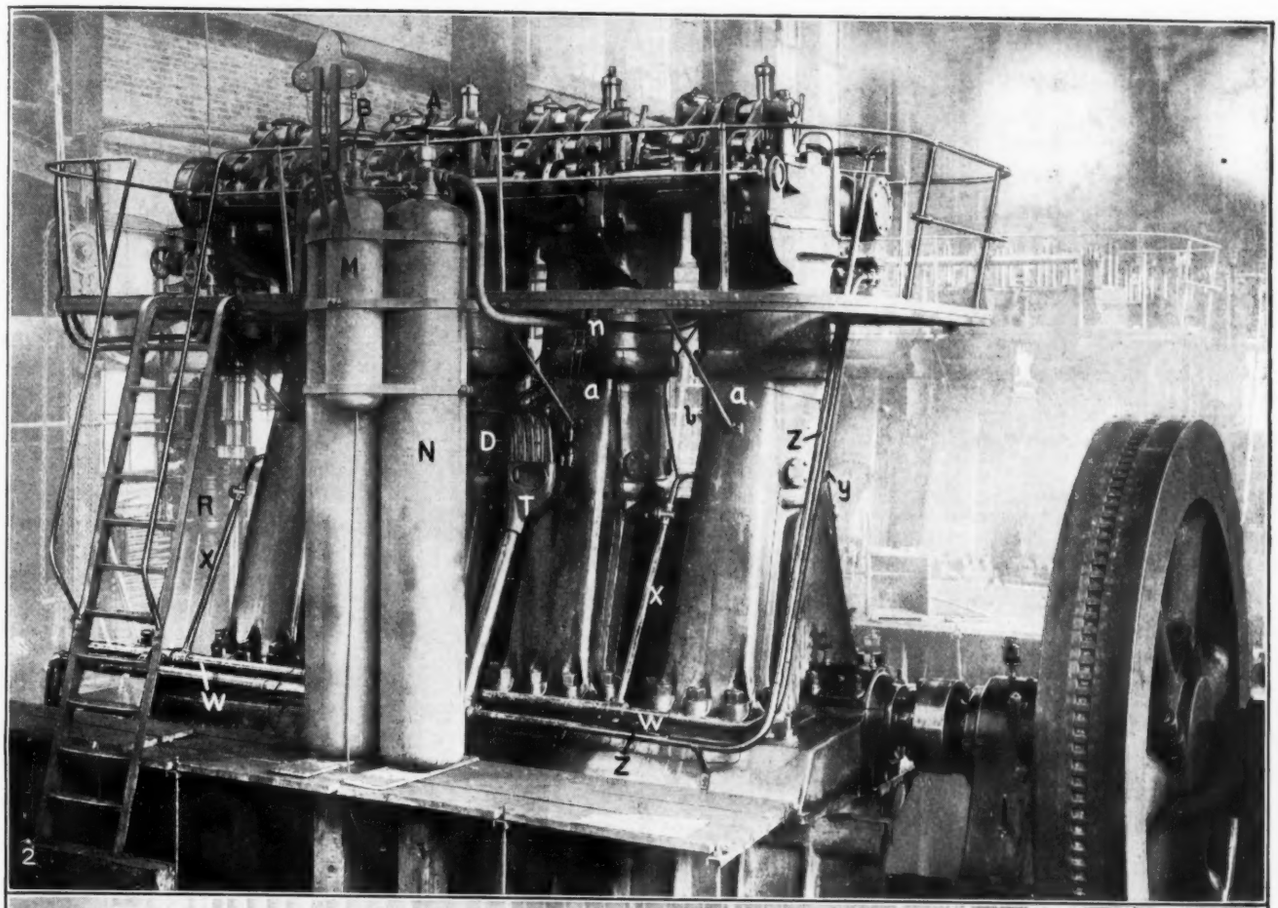
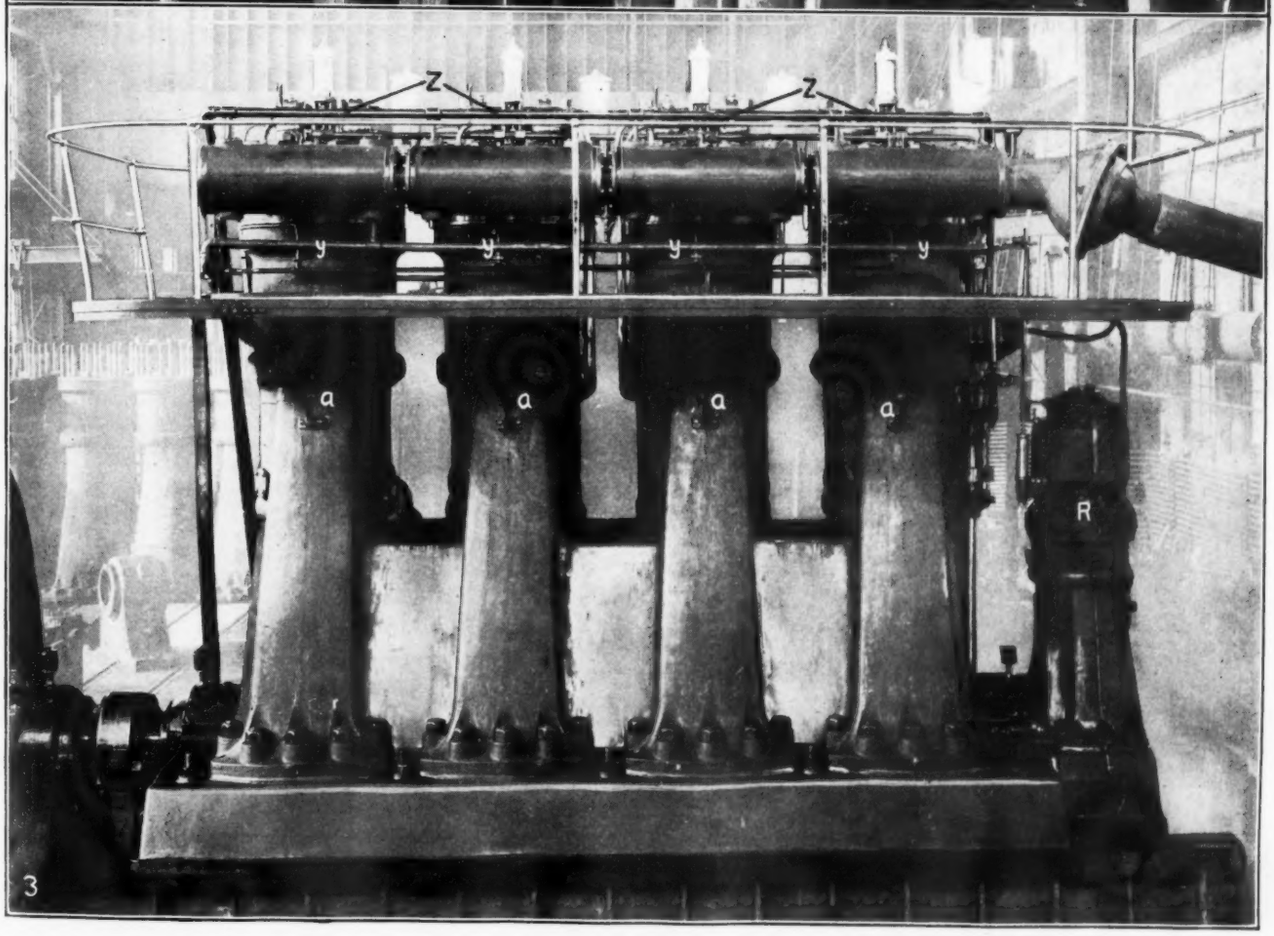


FIG. 1. PLAN AND ELEVATIONS OF 500-HP. McINTOSH & SEYMOUR DIESEL ENGINE, A-FRAME TYPE

In Fig. 4 *A* is the top of the air-inlet valve cage. The valve opens inwardly and admits air, which is drawn through the slotted intake *B* as the piston makes its downward stroke. *C* is the top of the exhaust-valve cage. This valve also opens inwardly and is held open by the hooked lever *D* when the engine is turned over by hand. The exhaust valve itself and the cage detached are shown in Fig. 11, where *A* is the renewable seat on the valve and *B* the water jacket for the stem. The valve is assembled by slipping the cage over the stem, putting the spring in place, compressing it until the slotted collar *C* can be slipped into the groove *D* and allowed to settle into the pocket *E* in the top of the spring cap. The air-admission valve is similar to the exhaust valve except that its stem guide, being in contact only with cool air, is not water-jacketed.

The exhaust valve is the part of a Diesel engine requiring the most frequent attention, since the hot gases impinge on it more in exhausting than on any other. Under the most unfavorable conditions, however, it ought not to be necessary to inspect, grind and clean this valve more than once in every six weeks. The process is very simple. The shaft *F*, Fig. 4, which carries the levers that operate the valves, has grooves near the ends, best seen in Fig. 9. The nuts *GG*, Figs. 4 and 9, are removed, the housing of the fuel valve unshipped and a sling, taking hold of the shaft at the grooved ends, lifts it free with its system of levers. Then, by removing the nuts *HH*, Fig. 4, and unmaking the jacket-water connections, the exhaust-valve can be lifted out, a spare inserted and the engine put into running condition again in a few minutes; the valve taken out being cleaned and put in order at one's leisure. The air and fuel valves require only infrequent attention, and a regular schedule of a valve once in a fortnight will keep a 4-





FIGS. 2 AND 3. FRONT AND REAR VIEWS OF 500-HP. McINTOSH & SEYMOUR DIESEL-TYPE ENGINE ON TEST BLOCK

cylinder engine in excellent condition, under ordinary circumstances. This, with the inspection and cleaning of each cylinder once in every 6 to 9 months, together with ordinary attention to the bearings, is all that the engine requires. This certainly compares favorably with boiler cleaning and furnace repairs, the packing of stuffing-boxes and the leaky pipe connections requiring attention in a steam plant. The comparison becomes much more marked in localities where the steam-plant troubles are increased by bad water.

One of the cylinders (in the engine illustrated the one at the right in Fig. 2 and at the left in Fig. 3) has an additional valve to admit high-pressure air for starting. The top of this valve cage is shown at *J* in Fig. 5, and the cam operating it by the same letter in Fig. 9.

The Process of Starting Up

When the engine is to be started, all of the exhaust valves are hooked up by the levers *D*, when, compression being avoided, the engine can be easily barred into a position where the piston of the starting cylinder is just commencing its downward working stroke. The exhaust-valve levers are then unhooked. Before doing this the lever *L*, Fig. 9, has been thrown to the right, carrying the roller *K* over to the right-hand side of the cam, where it will be raised by the projection *J* when the crank has passed the center and the cylinder is ready to take air. The valve *A*, Figs. 2 and 6, which controls the flow of air from the starting-air tank *N* to the cylinder through the pipe *n*, Fig. 2, is opened quickly, and air at 750 lb. pressure is admitted to the waiting piston. When the engine starts, the valve *B*, which controls the supply of air to the nozzles through which the oil is sprayed into the cylinders, is opened. As soon as the engine has made about a revolution under the action of this air, the heat generated by compression in the other cylinders is sufficient to ignite the charge of oil that is sprayed in at each working stroke, and when this happens the starting air is shut off by pulling the handle *L*, Fig. 9, to the left so that the roller *K* simply rides upon the concentric left-hand half of the cam and does not raise the valve. The large receiver beside *N* is simply a spare; the compressor delivers into the smaller tank *M*. When the pressure in either of the larger tanks has been lowered by using air from it for starting or by any other cause, the tanks are all put into communication by opening the valves *aab*, Fig. 6, and when they are equalized and the pressure is high enough, both the larger tanks are shut off, tightly bottling up enough air to start the engine many times. The valves on these steel bottles have special disks and seats and are so tight that those sent from Sweden with an engine imported by the McIntosh & Seymour Corporation when considering undertaking its manufacture lost little or no pressure during transit and were used to start the engine when first erected in their shops. The smaller tank *M* remains in action as storage between the compressor and the spray nozzles, its lesser volume being better adapted to variation of the air pressure in accordance with the load.

Fuel Injection and Governing

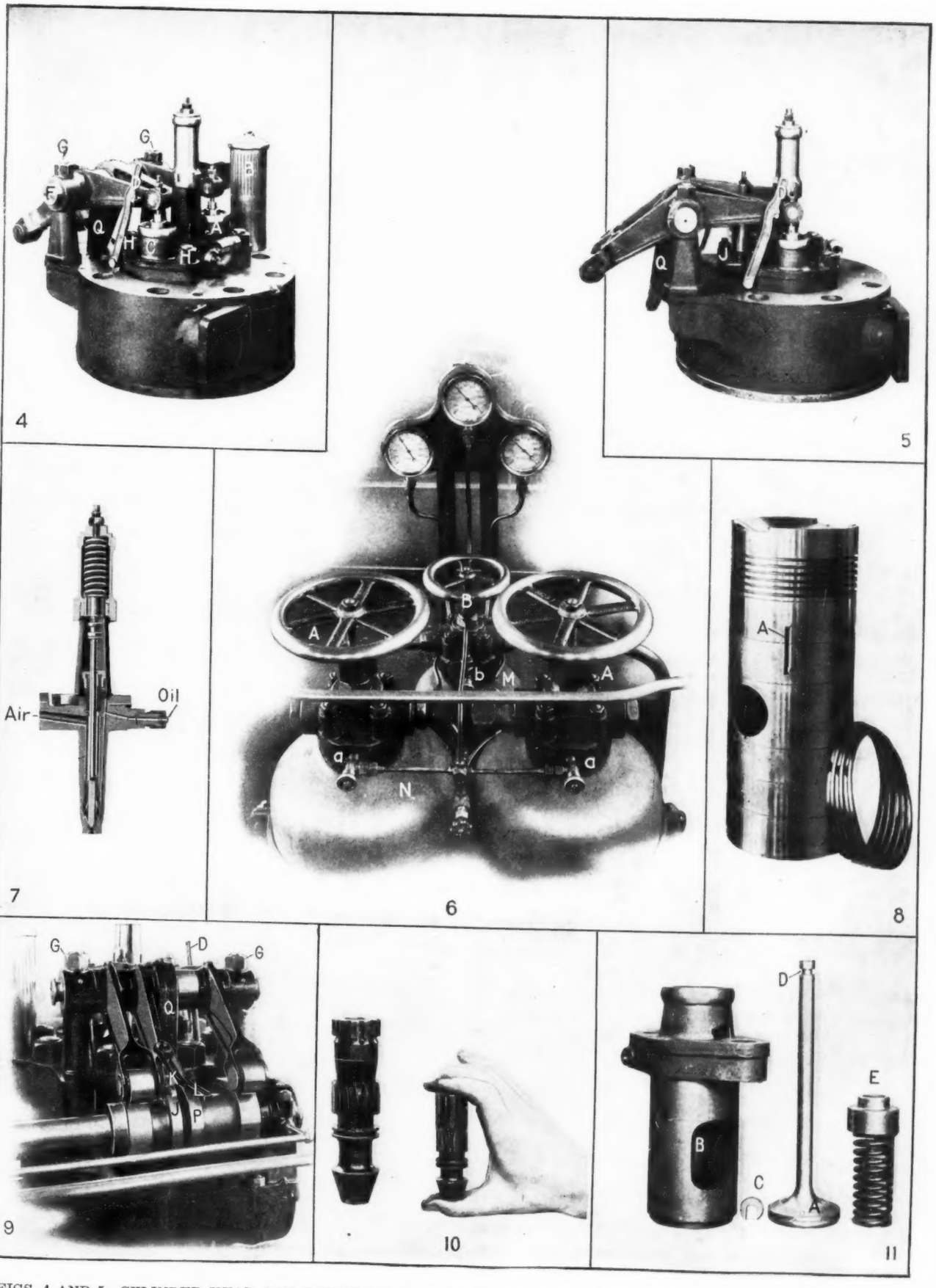
For each cylinder there is a separate fuel pump, the four pumps for the engine being assembled in pairs in two right-angled lines. Each pair of pumps is placed end to end, so that the four valves of each pair are fitted in two adjacent chambers in the middle of the pump barrel. In this position they are very accessible and can be removed if the gate that holds down the valve caps has been slacked and swung around. To drive both lines of pumps only a single eccentric is used, the plungers nearest the eccentric being operated directly, while those at the back ends are driven by return rods, a small cylindrical crosshead being fitted both back and front of each line of pumps. Actually, the eccentric that drives the fuel-pump plungers is floating upon another eccentric, and it is by the action of the governor in altering the relative positions of the two eccentrics that the stroke of the pump plungers is lengthened or shortened, thus producing the variations required in the fuel charges fed to the cylinders. The governor is mounted on the vertical shaft that drives the camshaft.

The fuel pump and governing mechanism represent the only departures in the McIntosh & Seymour engine from the practice of their Swedish associates, who employ the usual European system in governing, which regulates the amount of oil by spilling more or less of the delivery of the fuel pump back to the source of supply by holding the suction valve open for a greater or less period through the action of the governor. The object of this plan is to lighten the work that the governor has to perform and avoid any unfavorable influence upon its action by the reaction of the fuel pump.

The Governing Mechanism

On account of the long and successful operation on McIntosh & Seymour steam engines of shaft governors in which movable eccentrics, the positions of which are directly controlled by centrifugal weights with opposed springs, operate unbalanced gridiron steam valves having a thrust on the eccentric of many times that of the fuel-oil pump of an oil engine, it was decided desirable to adopt a somewhat similar system on the McIntosh & Seymour oil engines of Diesel type. In this the stroke of a separate pump plunger for each cylinder is varied directly by the governor by mechanism that prevents any undesirable reaction on the governor due to the thrust of the pump plungers. This plan requires a governor not materially larger than when the spilling system is used. There is also the advantage that if the governor mechanism becomes deranged with the spilling system it is liable to cause an excess of oil to be fed to the engine, resulting in a dangerous increase of speed, whereas with the McIntosh & Seymour plan the result would be simply to stop the engine.

The atomizer used on the McIntosh & Seymour engines is that invented by K. H. E. Hesselmann, of the Aktiebolaget Diesels Motorer. This atomizer has been tried out with various American oils and gives smokeless exhaust and perfect combustion under a wide range of loads, even with Mexican oils as heavy as 14 deg. Baumé. With ordinary oils the excess of atomizer-air pressure above compression pressure runs less than 300 lb. Its essential feature is that, instead of pulverizing the oil by crowding it down through perforated plates, it draws it by the injector principle into the current of ingoing injection air, which atomizes and absorbs it as fast as it is drawn up. In Fig. 13 the charge of oil is deposited in chamber *C*, which it does not fill, even at an overload. Air is admitted to the chamber *A* and passes through the ports *B*, while it has also access to chamber *C* through the space *EE*.



FIGS. 4 AND 5. CYLINDER HEAD AND VALVE MECHANISM. FIG. 6. AIR BOTTLES AND CONTROL VALVES. FIG. 7. THE FUEL ATOMIZER. FIG. 8. THE TRUNK PISTON. FIG. 9. THE CAM SHAFT AND VALVE OPERATING LEVERS. FIG. 10. DETAILS OF ATOMIZERS. FIG. 11. THE EXHAUST VALVE AND CAGE

When the fuel valve opens, the air, rushing through the port *BB*, passes through the expanding passage *F* between the valve stem and the receding wall of the surrounding fitting and induces by injector action a difference of pressure which causes the oil to flow to the space *J*, into which the oil, having been elevated and broken up through the slotted plates *K* and *L*, is drawn and picked up by the ingoing air. The form of the fuel plate *M* has an important effect upon the efficiency of the atomizer and varies according to the fuel.

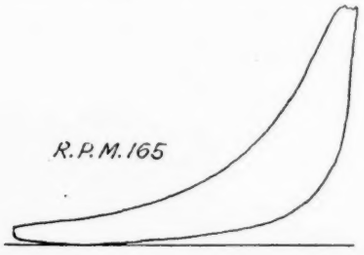


FIG. 12. DIAGRAM FROM DIESEL ENGINE

Each of the cylinders in the engine illustrated develops about 125 brake horsepower at rated load and takes about one-third of a cubic inch of oil per working stroke. Such a charge of oil is burned on a 4-cylinder, four-cycle engine once each stroke, or twice each revolution, so that the regulator in apportioning the fuel to the load acts with the same frequency as that of a single-cylinder double-acting steam engine. When it is considered that on tests instantaneous departure from the average of not more than 2½ per cent. and sustained departure of not more than 1½ per cent. from full load to no load are obtained in usual governing and that much closer regulation can be obtained, the sensitiveness of the governor controlling this small amount of fuel is apparent.

This regulated amount of oil is forced into the interstices of the spray nozzle, Fig. 7, whence it is ejected into the highly heated and compressed air in a finely atomized condition by the air from the tank *M*. This air has a pressure of 700 to 800 lb., varying according to the density and viscosity of the fuel, also according to the load conditions. As the compression pressure is not over 500 lb., there remains a pressure difference sufficient to produce a vigorous injecting and atomizing effect even through the reduced passages. The rate of entrance and combustion of the fuel, and thus the shape of what would be the steam line in a diagram from a steam engine, is dependent upon this pressure difference, which is modified for different load conditions by varying the pressure in the injection-air tank *M*. The fuel valve opens upward and is operated by the cam *P*, Fig. 9, the lever arm *Q*, Figs 4, 5 and 9, being upon the under side of the cam instead of above, as are the levers of the inwardly opening valves.

The air under pressure is supplied by a two-stage compressor with inter- and after-coolers, driven by a crank on the end of the main shaft and shown at *R* in Figs. 2 and 3 and in section in Fig. 1. The driving of this compressor requires about 5 per cent. of the power of the engine, but as all of this air is reëxpanded in the main engine cylinders, much of this energy gets back into the system.

The compression pressure attained depends upon the percentage of clearance by volume, much of which is in the dished head of the piston (see Fig. 8). The piston is of ample length to take the side thrust, which is kept normal by using a connecting-rod six cranks in length.

The piston is lubricated by forcing oil upon it between the rings when it is in its lowest position at the end of the working stroke. The force-feed oil pump is on one of the center frames, hidden by the air receivers in Fig. 2, but the points of delivery are shown at *aaa*, Figs. 2 and 3. It was currently supposed that this oil must be forced in against a high pressure, and the pumps and piping system are proportioned accordingly, but the engineers of the McIntosh & Seymour Corporation became curious and tested it by attaching a high-pressure gage to the pump delivery and found that the pressure was hardly enough to move the pointer away from the pin, which is natural when one comes to think of it, if the rings are tight.

The wristpin is lubricated by forcing oil into the pocket *A*, Fig. 8, whence it runs down to the pin by gravity and inertia. The point at which this oil is delivered on the right-hand cylinder is shown at *b* in Fig. 2.

The pipe *W* is for the incoming jacket water. Two of the connections to the cylinder jackets are shown at *XX*. The pipe *Y* connects to the similarly lettered pipe beneath the exhaust pipe in Fig. 3 and delivers water to the jackets of that pipe. The water, having passed through

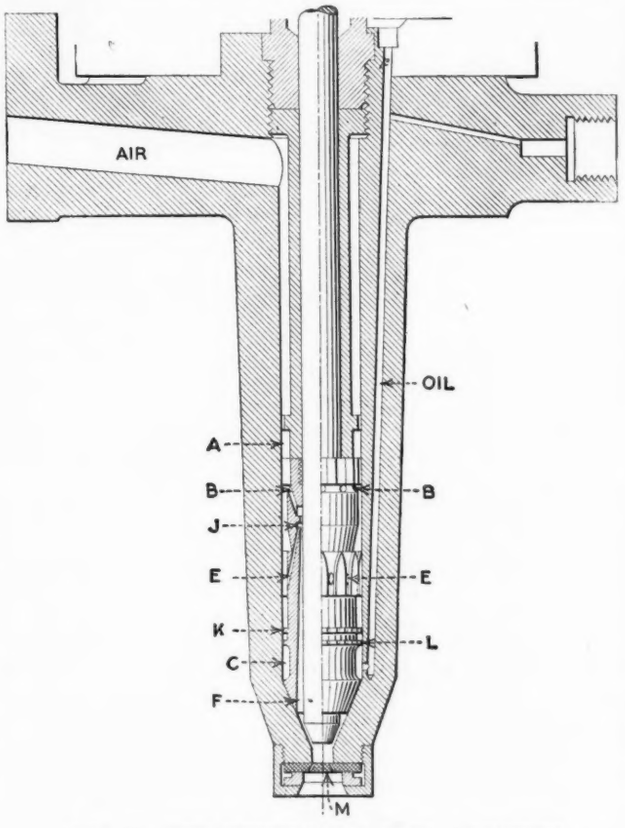


FIG. 13. THE HESSELMANN FUEL ATOMIZER

the jackets, is delivered to the pipe *ZZ* and conveyed to waste by the pipe *Z* in Fig. 2. The fixture *T* on the front of the engine, Fig. 2, is the receiver for the open deliveries for the jacket water from the four cylinders, the four exhaust-valve stems and the intermediate and after coolers of the compressor. The attendant can tell by passing his fingers along the opening if each is running at the right temperature.

The frame is made in two forms, the A-type shown in Figs. 2 and 3, and the B-type, which is shown in Fig. 14.

Even the joints where the frames go together are scraped and fitted, so that everything is square and in line when bolted up. The pig iron for the cylinders, pistons and other heavy parts that come in contact with the burning oil is imported from Sweden, and the smaller parts, such as valves, which require special machinery and do not involve heavy freight charges, are procured complete from the Swedish builders, of whom the McIntosh & Seymour Corporation is the American licensee.

The A type is made with 4 cylinders, all of 18⅞-in. diameter, and 28⅜-in. stroke, running at 165 r.p.m. and developing 500 b.hp. The B-type, Fig. 14, is made with from 1 to 6 cylinders running from 120 to 280 r.p.m. and

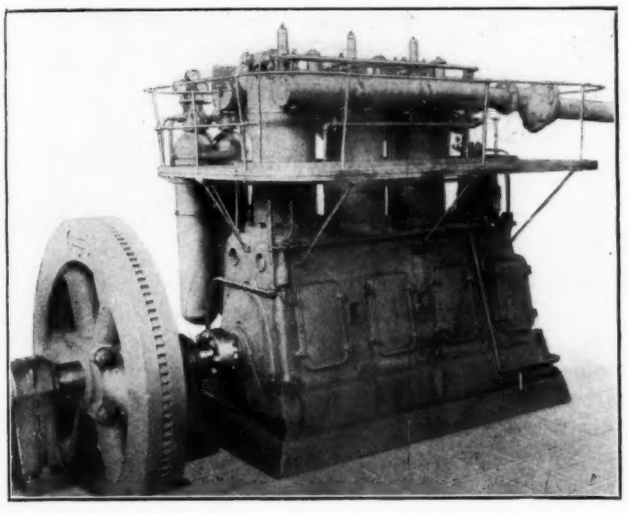


FIG. 14. McINTOSH & SEYMOUR DIESEL, TYPE B

developing from 40 to 1,000 b.hp. The following results from tests on a 4-cylinder A-type engine at the McIntosh & Seymour shops are furnished.

RESULT OF TEST ON TYPE A ENGINE

Rated horsepower					500
Cylinder diameter, in.					18⅞
Stroke, in.					28⅜
Rated speed, r.p.m.					164
Kind of fuel					Fuel oil
Load, per cent	110.5	100.2	77	52	26.2
R.p.m.	162	164	168.5	170	172
Brake horsepower	553	501	385	260	131
Duration of test-hours	0.5	1	0.7	0.67	0.5
Fuel consumption per brake horsepower-hr. in lbs.	0.387	0.393	0.398	0.418	0.565
Injection pressure, lbs.	900	850	800	820	790
Exhaust-gas appearance	Clear	Clear	Clear	Clear	Clear
Inlet temp. of cooling water, deg. F	70	70	70	70	70
Outlet temp. of cooling water, deg. F.	156	160	160	160	158
Room temp., deg. F.	76	76	75	76	76

Fig. 12 is a reproduction of an actual diagram from the engine. The indicated horsepower at the time of the above tests is not given but we have seen diagrams which interpreted under the given conditions of brake horsepower, spring, etc., indicate a mechanical efficiency of over 80 per cent.

Flashpoint Determinations have no definite connection with lubricating power. None of the lubricating oils on the market have a dangerously low flashpoint. For gas-engine cylinder lubrication a reasonably high flashpoint is necessary.

Testing for Flashpoint may be done in a simple manner by heating a few ounces of oil over a gas flame and trying a lighted match over the surface of the oil until a wave of flame is seen to flash across its surface. The temperature of the oil when this occurs is called its flashpoint. Oils whose specific gravities are below 0.85 generally have flashpoints below 60 deg. F., while those of which the specific gravities exceed 0.85 usually have higher flashpoints.

The Oil Economizer

The inventor of this device noticed that when cylinder oil was dropped upon the surface of heated water it became broken up into minute globules. At the exhibition in connection with the recent convention of the National

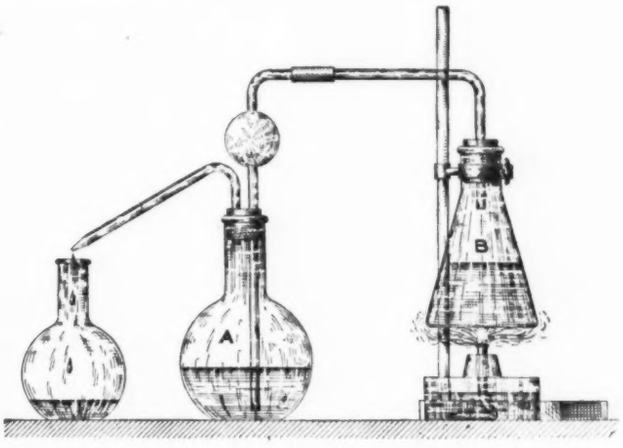
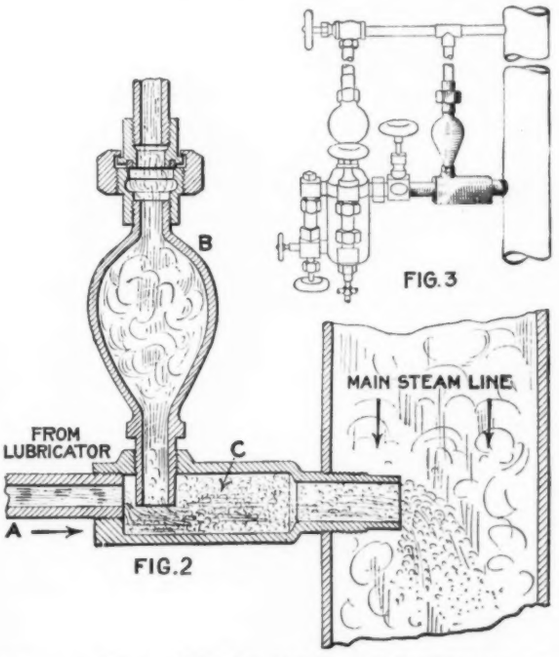


FIG. 1. EMULSIFYING CYLINDER OIL WITH STEAM

Association of Stationary Engineers at Columbus, its promoters performed the demonstration illustrated in Fig. 1.

A measured quantity of cylinder oil was placed in a beaker *A* and steam, generated in the flask *B*, was passed through it. In about a half an hour the volume of the contents of *A* was increased to some three times that of the measured quantity of oil with which the experiment started, and only after standing for hours would the water settle out.

It is not claimed that by this process one makes three gallons of oil out of one, but that the same quantity of



FIGS. 2 AND 3. SECTION AND APPLICATION OF OIL ECONOMIZER

oil in this greater volume can be more efficiently distributed and applied.

The device by which the process is effected in practice is shown in section in Fig. 2 and applied to a sight-

feed lubricator in Fig. 3. The oil coming from the lubricator at *A* meets the steam coming through the condenser *B* and is mingled with it in the mixing chamber *C*, passing to the steam main in the emulsified condition described. It will be interesting to see if this action obtains with straight mineral as well as with compounded oils and with oils having an asphaltic as well as paraffin base. The Oil Economizer Co., 616 Columbus Savings and Trust Building, Columbus, Ohio, is the manufacturer.

Calculating Dry Flue-Gas Loss

By Carleton W. Hubbard*

SYNOPSIS—Much of the matter published and in use regarding the relation of the carbon dioxide content to the dry flue-gas loss so confounds coal, carbon, fixed carbon, and the "amount of coal containing one pound of carbon," that varying results may be obtained from the same percentage of CO_2. The following method of determining that loss was worked out by the author for use in an engineering office dealing with combustion.

In this article the letters in the formulas represent the following:

P = Percentage of carbon dioxide in the dry flue gases.
C = Fractional amount of carbon in one pound of coal.
O = Fractional amount of oxygen in one pound of coal.
H = Fractional amount of hydrogen in one pound of coal.
H_a = Fractional amount of hydrogen in one pound of coal for which air must be supplied.
A_h = Cubic feet of air which must be supplied for the combustion of hydrogen.
N = Fractional amount of nitrogen in one pound of coal.
N_v = Volume of nitrogen in one pound of coal liberated when the coal is burned.
A_e = Amount of excess air supplied per pound of carbon or coal.
A = Amount of air supplied per pound of carbon.
V_d = Volume of the dry products of combustion per pound of carbon or coal.
W_d = Weight of the dry products of combustion per pound of carbon or coal.
L_d = Loss in B.t.u. due to the sensible heat of the dry flue gases.
T = Temperature of the flue gases at the point where they leave the boiler.
t = Temperature of the air in the boiler room.

Aside from its use as an indicator of furnace efficiency, the carbon-dioxide content is a necessity in computing the dry flue-gas loss, which is usually the largest single preventable loss in the boiler room. To compute this loss the weight of the dry products of combustion per pound of coal must be known, and also the number of degrees these gases must be raised in temperature. The only way of obtaining the weight of the dry products of combustion is by having a record of the carbon dioxide and a knowledge of the total carbon in a pound of the coal. If the coal is assumed to be all carbon, the dry flue-gas loss will be too large; while if the fixed carbon as shown by the ordinary proximate coal test is used as the value of the total carbon, the loss will be too small. That the total carbon and the average percentage of CO_2 are all that is needed to determine the weight of the dry products of combustion per pound of coal will be shown.

The process of the formation of carbon dioxide is represented by the following equation:

$$C + O_2 = CO_2$$

One atom of carbon (C) unites with two atoms of oxygen (O_2) to form one of carbon dioxide (CO_2). The atoms of oxygen are 2.66 times as heavy as the atom of carbon, and this means that each pound of carbon requires 2.66 lb. of oxygen to burn it completely. The product of this combustion is 3.66 lb. of carbon dioxide.

The oxygen for burning fuel in ordinary practice comes from the air, which by weight contains 23% of it, the remaining 77% being nitrogen. By volume the composition of the air is different, being 21% oxygen and 79% nitrogen. At 62 deg. F. and atmospheric pressure, 1 cu.ft. of air weighs 0.0761 lb. If 2.66 lb. of oxygen is required to burn 1 lb. of carbon it will take $\frac{2.66}{0.23} = 11.6$ lb. of air, or $\frac{11.6}{0.0761} = 152$ cu.ft. of air to supply this amount of oxygen. If 152 cu.ft. of air is used to burn 1 lb. of carbon, the resulting products of combustion will consist entirely of carbon dioxide and nitrogen. All of the oxygen will have united with the carbon, and the nitrogen will remain unchanged. At a given temperature one atom of carbon dioxide occupies the same space as one atom of oxygen, so that in these products of combustion there will be 21% of carbon dioxide and 79% of nitrogen. Suppose that twice as much air (304 cu.ft.) is supplied for each pound of carbon. The amount of carbon is the same as before, but there is twice as much oxygen. The result is that only one-half of the oxygen will be used and that the percentage of carbon dioxide will be half as great as before. Tabulating these two conditions gives:

Cubic Feet of Air per Lb. of Carbon	Products of Combustion			Volume of the Products of Combustion
(A)	CO_2	O_2	N_2	V_d
152	21.0	00.0	79.0	152
304	10.5	10.5	79.0	304

From this it is apparent that as the amount of air supplied increases, the percentage of carbon dioxide decreases. This relation can be stated in the form of an equation:

$$\frac{152}{A} = \frac{P}{21} \text{ or } P = \frac{152 \times 21}{A} \quad (1)$$

*Engineer, Fuel Engineering Co., New York City.

Since the products of combustion are equal in volume to the amount of air supplied, this equation can be written in another way, thus:

$$P = \frac{152 \times 21}{V_d} \qquad (2)$$

This equation is fundamental in all discussions of carbon dioxide, and when plotted gives what is called as the "pure-carbon curve." Under boilers commercial coal and not pure carbon is used, and if equation (2) is used on flue-gas analyses where coal is being burned, the results will naturally be in error. It is possible, however, to derive a formula from equation (2) that will be nearly correct for any given coal.

Before doing this it will be necessary to consider what factors in coal will affect the results. While the larger part of the combustible in the coal is carbon, it contains some hydrogen, oxygen and a small amount of nitrogen. Hydrogen produces more heat per pound than any other substance in the coal, while the nitrogen passes through the combustion process unchanged. The oxygen in the coal is assumed in this discussion to be used entirely to burn the hydrogen in the coal. There is not enough oxygen in the coal to burn all of the hydrogen, so that some additional air has to be supplied to burn the hydrogen, which forms water when it unites with oxygen. The process of the formation of water is represented as follows:

$$2\ H_2 + O_2 = 2\ H_2O$$

Two atoms of hydrogen unite with one of oxygen to form two of water. In this case the atoms of oxygen are eight times as heavy as those of hydrogen, which means that one pound of hydrogen requires 8 lb. of oxygen to burn it completely. The product of this combustion is 9 lb. of water. The air required to furnish 8 lb. of oxygen is $\frac{8}{0.23} = 34.8$ lb.; or $\frac{34.8}{0.0761} = 457$ cu.ft. The oxygen in the coal can be used to burn one-eighth its weight of hydrogen. The hydrogen for which air must be supplied is therefore not the total amount of hydrogen in the coal, but a lesser amount H_a, where

$$H_a = H - \frac{O}{8}$$

The number of cubic feet of air which must be supplied for the hydrogen in the coal is then

$$A_h = 457 \times H_a$$

The instrument which analyzes the flue gases determines the volume occupied by its different constituents. It has been shown that the burning of hydrogen results in the formation of water. When a sample of gas is taken from a boiler uptake this water is present in the form of superheated steam, which condenses before it gets to the analyzer. As a result of this the testing instrument does not register the amount of water formed, but it will be affected by the increased volume of the flue gases due to the nitrogen in the air which was supplied to burn the hydrogen. This will amount to $457 \times 0.79 = 361$ cu.ft. of nitrogen for each pound of hydrogen.

It now becomes necessary to derive an expression for the total products of combustion. The dry products of combustion are as follows: (a) The carbon in the coal; (b) the air supplied to burn the carbon; (c) the excess air; (d) the nitrogen from the coal; (e) the nitrogen from the air supplied to burn the hydrogen.

It should be noted that (a) and (b) do not appear in the flue gases in the form as shown in the foregoing, but as carbon dioxide and nitrogen. It has been shown that when carbon and oxygen unite to form carbon dioxide, the latter occupies the same space that the oxygen alone formerly required. Consequently the carbon in the coal does not increase the volume of the flue gases, and the sum of the volumes of (a) and (b) is equal to the volume of (b). The weight of the volume of gases is now changed, the volume (b) having a weight equal to the sum of the weights of (a) and (b).

The volume of the dry products of combustion from one pound of coal may therefore be expressed as follows:

$$V_d = 152 \times C + A_e + N_v + 361 \times H_a \qquad (3)$$

Equation (2) is written for one pound of carbon, and to adapt it to the use of coal it becomes necessary to multiply the numerator by the fractional amount of carbon in one pound of coal. Doing this and substituting the value of V_d just found gives:

$$P = \frac{152 \times C \times 21}{152 \times C + A_e + N_v + 361 \times H_a} \qquad (2)$$

By transposing this equation and solving for A_e it is possible to derive an expression for the volume of the excess air, thus:

$$A_e = \frac{152 \times C \times 21}{P} - 152 \times C - N_v - 361 \times H_a \qquad (2)$$

By substituting this value of A_e in equation (3), it is possible to simplify the expression for V_d, thus:

$$V_d = 152 \times C + \frac{152 \times C \times 21}{P} - 152 \times C - N_v$$

$$- 361 \times H_a + N_v + 361 \times H_a = \frac{152 \times C \times 21}{P} \qquad (3)$$

This value of the dry products of combustion is a correct volumetric expression. The percentage of carbon dioxide in the flue gases is determined by a machine that measures volumes and not weights. In combustion computations, however, the "pounds of dry combustion products per pound of coal" is the value usually desired. The only case in which the weights are not proportional to the volumes is where the carbon dioxide has taken the place of the oxygen in the air supplied for the combustion of the carbon and, as shown in equation (3), this happens to be the only factor that affects the final result. In other words, while the volume of the carbon has disappeared, its weight has not, and the resulting flue gases are heavier than the air supplied by the weight of the carbon. Equation (3) further shows that as far as the percentage of carbon dioxide is concerned, the volume of the dry products of combustion may be considered as equal to the volume of the air supplied for the combustion of the carbon, for the air supplied for the hydrogen, the excess air, and the nitrogen in the coal do not appear in the equation. The dry products of combustion will be air, carbon dioxide and nitrogen. Air and nitrogen weigh nearly the same per cubic foot, consequently if the dry products of combustion were considered to be air and carbon no serious error would be introduced. If, then, the volume of the dry products of combustion be divided by the weight of one cubic foot of air and the weight of the carbon per pound of coal be added, the re-

sult will be the weight of the dry products of combustion. This may be expressed as an equation, thus:

$$W_d = \frac{V_d}{0.0761} + C = \frac{152 \times C \times 21}{0.0761 \times P} + C = \frac{243.6 \times C}{P} + C \quad (4)$$

To raise one pound of the dry products of combustion one degee Fahrenheit requires 0.237 B.t.u., so that if W_d be multiplied by 0.237 and by the number of degrees by which the temperature of the flue gases exceeds that of the air entering the furnaces, the result will be the dry flue-gas loss in B.t.u. This can be expressed as a formula, thus:

$$L_d = W_d \times 0.237 \ (T - t) \quad (5)$$

To summarize the subject the following example will be taken: B.t.u. in coal as fired, 13,608; total carbon in coal as fired, 77.13 per cent.; fixed carbon in coal as fired, 66.12 per cent.; average CO_2, 11.25 per cent.; average stack temperature, 470 deg. F., and average temperature in the fire room, 70 deg. F.

$$W_d = \frac{243.6 \times 0.7713}{11.25} + 0.7713 = 17.46$$

$$L_d = 17.46 \times 0.237 \ (470 - 70) = 1{,}655 \ B.t.u.$$

The percentage loss in the dry flue gases is therefore

$$\frac{1{,}655 \times 100}{13{,}608} = 12.16 \ per \ cent.$$

That the errors involved by using the fixed carbon as the total carbon or by assuming the coal to be all carbon may be avoided, the foregoing example will be worked out in two ways—first, assuming that the coal is all carbon and, second, assuming that the fixed carbon is the total carbon.

Assuming that the coal is all carbon,

$$W_d = \frac{243.6 \times 100}{11.25} + 100 = 22.63$$

$$L_d = 22.63 \times 0.237 \ (470 - 70) = 2{,}145 \ B.t.u.$$

Assuming the total carbon to be the fixed carbon,

$$W_d = \frac{243.6 \times 0.6612}{11.25} + 0.6612 = 14.97$$

$$L_d = 14.97 \times 0.237 \ (470 - 70) = 1{,}420 \ B.t.u.$$

By assuming the coal to be all carbon the dry flue-gas loss is 2,145 — 1,655 = 490 B.t.u. too large, and by using the fixed carbon for the total carbon it is 1,655 — 1,420 = 235 B.t.u. too small. The magnitude of the errors involved will be:

Error in assuming the coal to be all carbon,

$$\frac{490 \times 100}{1{,}655} = 29.6 \ per \ cent.$$

Error in assuming the fixed carbon to be the total carbon,

$$\frac{235 \times 100}{1{,}655} = 14.2 \ per \ cent.$$

Torque Characteristics of Direct-Current Motors

By Alan M. Bennett

SYNOPSIS—What the torque of a motor is, how it is determined, and how it varies in shunt, compound and series motors.

The function of an electric motor is to produce mechanical power by the application of electric energy to the motor terminals. In the commercial motor this is accomplished by rotation of the armature within the field or stationary part of the motor. The turning effort exerted on the armature, tending to cause it to rotate, is known as the torque of the motor, and inasmuch as torque must exist in order that rotation of the armature take place and power be developed, the function of a motor is often defined as the production of torque.

The ability of a motor to develop torque depends on the principle that a conductor carrying current, placed in a magnetic field at right angles to the lines of force, will tend to move across the field in a direction that will cause it to cut these lines. This motion is produced by the action between the lines of force of the field and those set up around the wire carrying current, these forces being proportional to the current and the strength of the magnetic field. In the motor, therefore, torque is dependent on the armature current and the flux density, and it exists when these two factors are present, whether the armature rotates or not. If the armature be held from turning, torque is still developed, though it may not be sufficient to cause rotation of the armature against the force holding it.

Torque is usually expressed in pound-feet, meaning the pull in pounds exerted at a radius of one foot. Therefore the actual pull in pounds exerted on the conductors at the surface of the armature must be multiplied by the radius of the armature in feet, to obtain the equivalent torque value. The work done by the motor in foot-pounds per minute equals the horsepower developed multiplied by 33,000, all of which is transmitted by the conductors on the armature surface to the shaft. By dividing this work in foot-pounds by the distance through which the armature surface moves in a minute, the total pull on the conductors in pounds is obtained. In one revolution any point on the armature surface passes through $2\pi r$ ft., where r equals the radius of the armature in feet; and in N revolutions per minute passes through $2\pi rN$ ft. per min. The total pull on the armature conductors therefore equals $\frac{HP \times 33{,}000}{2\pi rN}$ pounds, and the torque $T = \frac{HP \times 33{,}000}{2\pi rN} \times r = \frac{HP \times 33{,}000}{2\pi N}$ pound-feet. . . . . (1)

Rearranging formula (1), we have, $HP = \frac{2\pi TN}{33{,}000}$ . . . . (2), which is the expression for the mechanical horsepower of the motor in terms of torque and speed.

But that part of the electrical input to the motor converted into horsepower output equals $\frac{eI}{746}$ . . . . (3), where e is the counter electromotive force generated by the armature, and I is the armature current. As formulas (2) and (3) are both a measure of the work imparted to the motor shaft, they may be equated, or $\frac{2\pi TN}{33{,}000} = \frac{eI}{746}$, from which, $e = \frac{2\pi TN746}{33{,}000\,I}$ . . . . (4). The counter electromotive force also equals $\frac{\phi CNP}{10^8 \times 60}$ . . . . (5), this being the fundamental equation for the voltage generated by an armature, where ϕ = flux per pole; C = conductors per circuit on armature; N = r.p.m., and P = number of poles. Equating these two values $\frac{2\pi TN746}{33{,}000\,I} = \frac{\phi CNP}{10^8 \times 60}$, from which, $T = \frac{0.117\,\phi ICP}{10^8}$ . . . . (6).

From this equation may be determined the torque developed by the motor armature under any condition, whether standing still, at the time of starting or when rotating at any speed. The factors C and P having fixed

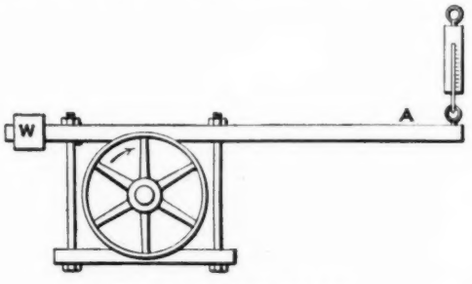


FIG. 1. PRONY BRAKE

values, it is seen that the torque varies directly as the current taken by the motor and the flux density, and also that it is independent of speed. The torque given by this formula, as well as that given by formula (1), is the total torque exerted on the armature, tending to cause it to rotate. The effective torque, or that exerted at the pulley, equals the apparent torque less that necessary to overcome friction, windage and core loss. The pull at the rim of the pulley, or the difference in tension on the two sides of the belt, equals this effective torque divided by the radius of the pulley in feet.

Measuring the Torque of a Motor

The torque developed by a motor may be experimentally determined by means of the prony brake. Referring to Fig. 1, the weight W counterbalances the arm A, so that when the motor is standing still there is no pull on the scales and the arm is horizontal. The jaws are clamped on the pulley till the motor takes the current at which it is desired to measure the torque. It is evident that the friction between the clamp and the pulley will cause a downward movement of the arm A and that a corresponding pull will be registered on the scales, depending on the degree of tightness with which the pulley is clamped. The torque, or turning effort, is represented by the product of the friction force on the pulley rim and the radius of the pulley in feet. If p is the pull in pounds registered on the scales and l is the length of the arm A in feet to the center of the pulley, then $p \times l$ must be the force in pound-feet balancing the turning effort, and a measure of the torque developed. In this case, if the motor is allowed to turn, the work done manifests itself in friction at the pulley rim and is dissipated in heat. If the clamp is tightened sufficiently to prevent the pulley from turning, pressure will still be registered on the scales and torque developed, but much in excess of normal, owing to the absence of counter electromotive force and the consequent heavy current. No work will be performed, and the energy applied to the motor will expend itself in heating the armature.

With a given friction force at the pulley rim it is evident that the pull on the scales will vary with the length of the arm A, increasing in exact proportion to the decrease in arm length; but as the torque is measured by the product of the pull and the length of the arm, the latter can have any length without affecting the measure of the torque. With the torque determined as stated and the speed of the armature taken by a counter, the horsepower developed can be calculated from formula (2).

Motor Takes Only Current Needed

It is characteristic of an electric motor that under any condition of load and speed it takes only the amount of current necessary, in connection with the flux, to develop the torque required to do its work at that speed. The manner in which the motor current is thus automatically suited to the load is as follows: The rotation of the armature and the cutting of the flux by the conductors generates a voltage, just as in the armature of a generator that is being driven by an external source of power. This voltage, known as the counter electromotive force, is opposed to that impressed at the motor terminals and limits the current that would otherwise flow through the armature. The greater the speed of the armature, the greater will be the counter electromotive force, as can be seen from formula (5). The amount of armature current will therefore depend on the speed of the armature, being a maximum when the armature is standing still, in which case it is limited only by the armature resistance, or $I = \frac{E}{R}$, where E is the impressed voltage and R is the resistance of the armature and brush contact. If the armature is allowed to revolve and a counter electromotive force to be generated, the current flowing will be that given by the formula $I = \frac{E - e}{R}$ . . . . (7), where e is the counter electromotive force.

With the motor running free the torque required is light, and the motor will run at such a speed that the counter electromotive force will allow only that current to flow, which, in connection with the flux, will develop the torque required at that speed. As load is applied to the motor, the torque required increases. The current therefore must increase, and to allow this the counter electromotive force must decrease, which is done by a lowering of the motor speed till the current passing is sufficient to generate the torque required at that speed. Again, if while running at a given speed, the load on the motor is decreased, it is evident that the torque required will decrease and with it the current necessary to produce this torque. The excess of torque existing will cause an increase of speed, thus increasing the counter

electromotive force and reducing the current to the required value. The current is thus automatically limited to that required by the motor to do its work under any condition of load.

The division of direct-current motors with respect to their design features into the three general types—shunt, series and compound—gives rise to a difference in the torque characteristics of these types. The preceding formulas give a means for comparing these characteristics and determining the types best suited to the various applications in driving machinery.

Torque Characteristics of Shunt Motor

The shunt motor with its field excited from a constant voltage will have a fixed value of field current, after reaching its final temperature, provided the applied voltage does not vary. The flux established by this current will therefore be constant and will remain so whether the motor is loaded or not. We have seen that the torque developed by the armature varies directly as the flux density and the motor current. Therefore, in the case of the shunt motor with its constant flux, the torque will always vary directly as the current taken by the motor. Commercial starters are usually designed to limit the current taken by the motor at the time of starting to a value approximately 50 per cent. in excess of that taken at rated full load. With this limitation, the torque of a shunt motor cannot exceed 150 per cent. of its full-load running torque, and unless the magnetic circuit is so designed that it is well saturated, will not reach this value, owing to field distortion by armature reaction, which occurs particularly on overload. The shunt motor therefore is not applicable to those cases where heavy loads have to be started and accelerated rapidly. Owing to its constant field strength, it is subject to but slight changes of speed over the range from no load to full load and is admirably adapted to those cases where practically constant speed is required and where the starting duty is not severe.

Fig. 2 shows the speed-torque curves of a shunt motor. The no-load current is shown by *o-c*, the corresponding torque by *o-a*, and the speed by *o-d*. As the motor is loaded, the torque increases practically in a straight line till full load is reached. Beyond this point the curve droops, owing to the effect of armature reaction. The practically constant speed, the feature that distinguishes the shunt motor, is shown by the speed curve, the change between these limits being approximately 5 per cent.

Controlling Speed of Shunt Motor

The speed of the shunt motor may be increased above normal by weakening the field. This is accomplished by inserting resistance in series with the field by means of a rheostat and regulating the same to obtain the speed desired. In this case the addition of resistance reduces the field current, and consequently the flux. The torque will therefore be reduced in exact ratio to the reduction in flux density, and as the speed is increased in this same ratio, the torque will decrease inversely as the speed. As the work performed varies directly as the product of the torque and speed, the horsepower developed will remain constant through the whole range of speed control.

An increase of speed by this means is brought about as follows: As the flux is decreased, the motor generates less counter electromotive force and takes more current. This occasions an increase of torque and causes the motor to accelerate its speed and to again increase its counter electromotive force; and this continues till the counter electromotive force reaches the point where it limits the current to such a value as will produce no further acceleration. The speed can thus be increased till the weakened field causes excessive sparking, beyond which it is not practicable to go. With the ordinary shunt motor this limit is reached at about 50 per cent.

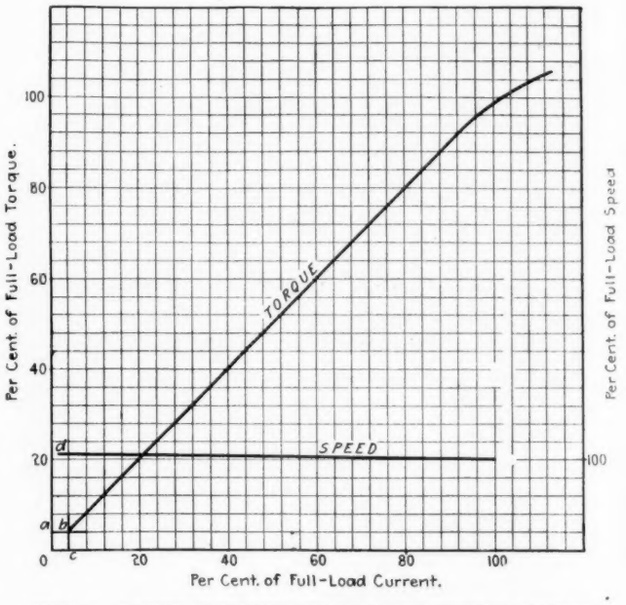


FIG. 2. SPEED-TORQUE CURVES OF SHUNT MOTOR

above normal speed, but with an interpole motor, it is possible to obtain speeds as high as five or six times normal, without injurious sparking. As the torque produced by the armature decreases as the field is weakened, it can be seen that a limit may be reached beyond which the torque may not be sufficient to maintain rotation, in which case the motor will come to a standstill.

The speed of a motor may be decreased below normal by inserting resistance in the armature circuit, thus varying the voltage impressed at the motor terminals. Speed adjustment by this means depends on the fact that the impressed voltage equals the counter electromotive force generated by the motor, plus the voltage drop in the armature circuit. As the impressed voltage is lowered, the counter electromotive force must correspondingly lower, in order that a given current may pass, and this can only take place by a decrease in speed. As the function of the resistance is merely to cause a drop of a certain portion of the line voltage, it is evident that the resistance can be designed to do this at full-load current, and that full rated torque can be developed through any range of speed that may be obtained by this means. Torque remaining constant, the horsepower developed will vary directly as the speed. This method of speed control, while convenient in many cases and possessing the advantage of constant torque through the whole range of control, is very inefficient, particularly at the lower speeds. The full working current is carried by the resistance, and the watts loss in the resistance is directly proportional to the voltage drop therein.

To meet the demand of those cases where a stronger torque is required than the shunt motor is capable of developing, the compound motor is used. The series field winding being in circuit with the armature carries the armature current, with the result that the field excitation is increased with the load, the flux is strengthened and the ability of the motor to produce torque is increased in proportion. The degree of compounding—that is, the ratio of series to the shunt ampere-turns—should be proportioned to suit the character of the service under which the motor operates. In many applications a few series turns will suffice to produce the necessary torque, while in others the excitation produced by the series field is a large percentage of the whole. In some cases, as in elevator service, where the required starting torque is heavy and the running torque light, the series turns on the field are cut out of operation by the starter or controller after the load has been started, and the motor is allowed to run on the shunt field alone. Rapid acceleration is thus obtained at starting, and constant speed independent of load when running. The efficiency is increased by this means, and speed adjustment by field weakening is more easily obtained. The change in speed from no load to full load will be greater in the case of the compound than the shunt motor, because of the increase in flux as load is taken by the motor, and this change will be greater the more heavily the motor is compounded.

Speed adjustment in the case of compound motors, by either field weakening or armature resistance, results in torque characteristics similar to those possessed by the shunt motor.

The compound motor described has the series turns on its field connected so as to assist the shunt field as load

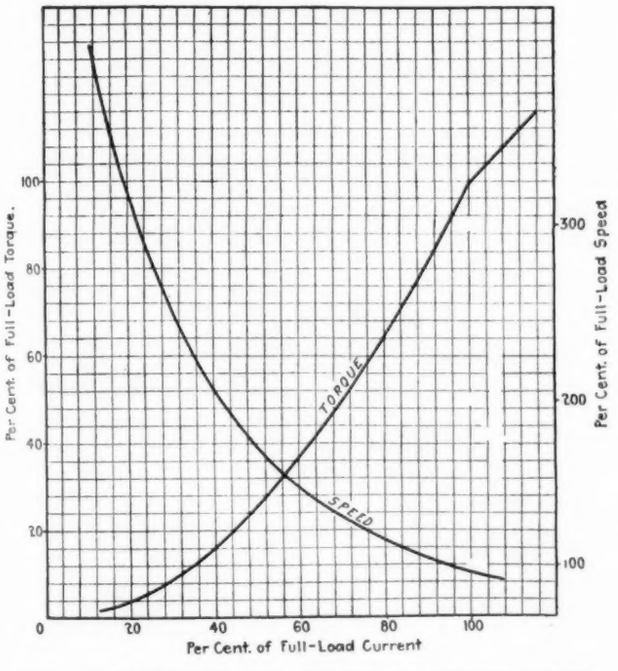


FIG. 3. SPEED-TORQUE CURVES OF SERIES MOTOR

is taken by the motor, and thus obtain some of the characteristics of the series motor. In the differential compound-wound motor the series turns are connected to oppose the shunt, with the effect that the field is weakened and the speed increased as the motor is loaded, the object being to compensate for the decrease in speed caused by voltage drop in the armature. The effect on the torque developed by a motor having this type of field is apparent. Maximum flux exists at no load and decreases as load is applied. At starting, therefore, the motor will be deficient in torque, and if under load, it either may not start or may start in the wrong direction, owing to the series field overpowering the shunt. If overloaded, the torque may not be sufficient to maintain rotation, in which case the armature current will rise to a dangerous limit because of the decrease in counter electromotive force. Owing to these bad features, which more than offset the advantage of constant speed obtained, the differential motor is not much used.

Torque Characteristics of Series Motor

The series motor is applicable to such classes of work as are found in crane, hoist, and traction service, where frequent starting at heavy load is encountered. Rapid acceleration in such cases is necessary, and these conditions can be met only with a very strong torque. In such applications considerable variation in speed under changing load conditions is not objectionable, and the series motor fulfills these conditions admirably. The field, being excited altogether by the armature current, will have a strength dependent on the load of the motor. Below the point of saturation of the magnetic circuit the flux will be proportional to the current, and the torque developed will therefore be proportional to the square of the current. As saturation is reached, which occurs at approximately full load and above this point, the flux becomes more nearly constant and the torque will gradually fall off till it is proportional to the current.

In Fig. 3 are shown the speed-torque curves of a series motor. Inspection shows that from the minimum load point the torque rapidly increases till approximately full load is reached. Beyond this point the increase is not so rapid, but there is not a falling off in torque due to the effect of armature reaction, as in the case of the shunt motor. The curves show that maximum torque will be had at minimum speed, which means maximum torque at starting. The speed of the series motor varies greatly with change of load, owing to the change in flux. Maximum speed exists at minimum load, and if the load is removed altogether or reduced to a low limit, the motor will race, the speed going beyond the limit of safety.

Controlling Speed of Series Motor

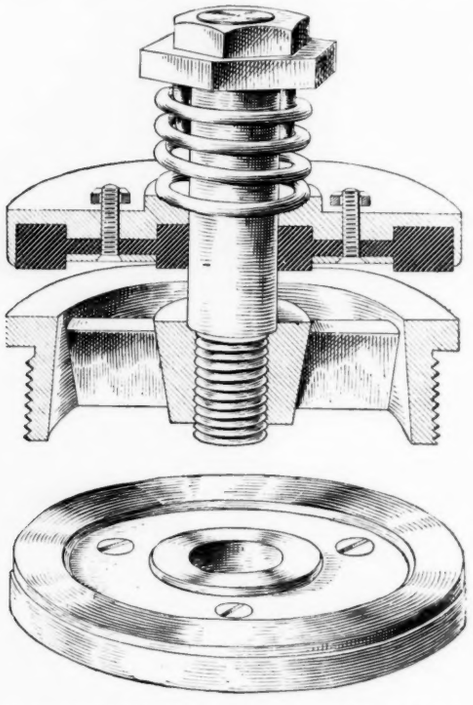
The speed of series motors may be controlled by resistance in the armature circuit, the method being similar to that employed with shunt motors, and the resistance may be designed to give constant torque through any desired speed range. Speed control by field weakening is accomplished by shunting a portion of the motor current through a path in parallel with the field, which has the effect of weakening the flux, the torque decreasing in the same proportion.

The foregoing will show that torque is an important factor, both in the design of the motor and in the application of the various types to the driving of machinery. It represents the ability of the motor to start and accelerate its connected load and maintain rotation at a given speed. Up to certain limits the torque may be increased by allowing the motor to take more current, but

this is often objectionable and recourse is had to means whereby stronger torque and more rapid acceleration may be had without causing an abnormal demand for current by the motor. This is accomplished by using the motor current to strengthen the field flux, series turns being wound on the field for this purpose, resulting in the compound and series motors having torque characteristics varying with the percentage of series turns used.

Rothenbucher Reversible Pump Valve

This reversible pump valve consists of but two parts, the aluminum cap and the rubber disk. When the face of the rubber disk becomes worn it can be reversed and the opposite surface used to face against the seat. The



DETAILS OF REVERSIBLE PUMP VALVE

illustration shows the construction of the valve. It is made in different sizes and adapted to various designs of pumps by the Coronum Equipment Co., 110 Fulton St., New York City.

Great Leaders have come indiscriminately alike from humble birth and from high rank. Remember Newton, the farmer's boy; Brahe, the Dane of noble family; Pasteur, the tanner's son. The father of Kepler was a soldier of fortune and his mother uneducated and undisciplined. Recall Faraday, the son of a blacksmith and himself a bookbinder's apprentice; Descartes and Napier, both of high rank and education; Hertz, the student of engineering in Munich; Helmholtz, the son of a professor and himself originally a surgeon in the Prussian army; Virchow, the son of a small farmer, while Darwin's grandfather was a famed botanist and his father a physician. Neither royal blood nor heredity have been proved as dominant factors. Leadership has defied country and ancestry, and environment. It is a God-given gift. So we have a right to expect of our schools leaders for the future of Science, but they will come, as they have all through the past, from the farm, from the city, from the poor, from the rich, from the high born, from the humble. The gift of the master mind knows no country, no ancestry, no environment.—From address by Lucien Ira Blake, Ph. D., Colorado School of Mines.

Power-Plant Equipment of the Massachusetts "Tech"

The new plant of the Massachusetts Institute of Technology will for the present develop 2,000 hp. It is unusually condensed, being ten feet narrower than any other station of like capacity. This is because the building lies in the 90-ft. strip belonging to Technology between Vassar St. and the railway. The latter affords economical delivery of coal in large quantities, and with mechanical stokers and other appliances the fuel is not handled by men at any stage of its use.

The boilers lie transverse to the street and railway to make more easy the addition of other boilers as the future needs of the growing Institute demand. There will be 1,560 hp. of new boilers and 400 hp. removed from Trinity Place, the new ones being Babcock & Wilcox boilers, with Riley stokers. The boilers will be equipped for heavy overloading.

There are features in this plant that make it different from ordinary commercial boiler rooms. It will be a laboratory for the use of students and will care for lighting, heating and power. The laboratory use is likely to demand a high peak load, which the establishment must be able to care for, but at the same time the load factor is comparatively low. The situation of the plant, besides affording the convenience and economy of direct coal delivery, is sufficiently removed from the educational and student portions of the Technology assemblage of structures to be no nuisance. The boilers are of the wrought-steel header type, designed for 175 lb. working pressure; the steam will be superheated 100 deg. F. The stack is 10 ft. diameter inside and 18 ft. outside and when the capping is placed, will stand 180 ft. above its base.

To convey steam to the buildings and laboratories where it will be needed, a reinforced-concrete subway, about 7 ft. square in inside measurement and 625 ft. long, is being built. It is to connect with the educational group through the administration building, and from this the steam will be distributed to the various points. For the present the supply will pass through a 20-in. pipe for utilitarian purposes and a 10-in. pipe will furnish the laboratories; there will be a 5-in. return for condensate. For furnishing condensing water for the turbines a concrete main has been laid. This conduit is 30 in. diameter, running back from the Charles River basin along the western edge of the great court. It will be more than a quarter of a mile long, and in rainy weather will collect the rainfall of the grounds and turn it in for condenser water.

The electrical outfit is an interesting one, since the engine room will serve for substation as well as for central station. The initial equipment will include three turbines direct-connected to generators—one of 750 kw., a second of 500 kw. and the third of 150 kw., furnishing three-phase current at 2,300 volts. There will then be one 150-kw. turbine and two 150-kw. motor-generators furnishing direct current at 110 and 220 volts, and two 35-kw. exciters, one steam- and one turbine-driven.

The steam laboratory is to be located in a long building near and parallel to Massachusetts Ave. High-pressure steam coming through the subway will be distributed to two mains below the ceiling of the first floor. There will

be a superheater here, connected at will with the machines where experiments are to be made. The condensers of the various engines in this laboratory are in the basement and take the cooling water from one of the large hydraulic canals, passing the warmed water into a hot-water return. In the basement there will be apparatus for determining the flow of superheated steam through orifices or turbine nozzles.

On the first floor of this building there will be located the engines, which will make a formidable showing. There will be a Curtis turbine of 75 kw. capacity, a 30-hp. Corliss engine with dynamometer attached to the flywheel, a 225-hp. McEwen tandem-compound engine, a compound engine and a generator of the same make of 250 hp., and a triple-expansion Corliss engine. These engines will be ranged along the western wall of the building. On the opposite side of the laboratory will be a Brown engine driving a three-stage compressor, which compresses air to 2,500 lb. per sq.in. There will be two or three other compressors and a few small engines of different types. A large power crane runs the length of this laboratory.

The Massachusetts Ammonia Safety Regulations

By Frank L. Fairbanks*

SYNOPSIS—A safety valve releasing ammonia into the atmosphere must be slow in opening to give ample warning of excessive pressure and should close slowly to avoid hammering the seat. A valve acting as a Napier orifice (one in which the ammonia escapes slowly until the pressure on the discharge side of the valve equals 0.585 of the initial pressure, then the more rapidly the back-pressure increases the wider the valve will open) has been invented to comply with the new ammonia regulations. Another requirement of the regulations is that the ammonia be discharged into the air through a diffuser. Emergency instructions in cases of ammonia leaks and inhalation are given.

The Massachusetts Legislature enacted on May 2, 1914 a law requiring a safety valve on every ammonia compressor and specifying that the regulations for governing the construction and application of these valves be formulated by the Board of Boiler Rules, and that the enforcement of the regulations be under the supervision of the Boiler Inspection Department. The writer was requested by the board to assist in preparing a code in compliance with the law, and after a conference it was decided to make a practical test of the different valves on the market, with the view of determining their merits and capacities as well as their defects.

The writer offered the facilities of the Sargents Wharf power house of the Quincy Market Cold Storage and Warehouse Co., for such tests, and suggested that Professor Miller, of the Massachusetts Institute of Technology, and Professor Read, of the Worcester Polytechnic Institute, be asked to form with me, a committee of three to go into the matter thoroughly and report its conclusions to the board. Preparations were made for not only a thorough test of well-known makes of valves, but for determining the flow of ammonia through orifices and the resistance to flow in pipe lines, etc.

These tests extended over a period of more than six months and were accomplished by pumping the ammonia gas from three 200-ton shell coolers with a 400-ton compressor discharging the gas through a 6-in. header to the suction side of a 1,000-ton compressor, by which means it was possible to get any pressure desired, as well as to get a sufficient degree of superheat to insure having a gas temperature so high that its character, while under observation, would be constant and without danger of a near approach to the saturated point. The entrance to this header was through a calibrated Napier's orifice, and it was provided with valves, test gages, oil wells for test thermometers, outlets and bypasses for attaching the safety valves to be tested. Six different ammonia safety valves of as many well-known makes were tested under the different conditions and under a wide variation of pressures.

The conclusions drawn from these tests were as follows:

That iron or steel valves with iron or steel seats were not safe, for the reason that the disk is readily corroded to its seat by a small amount of moisture, two of the six valves tested failing to open at 350 to 375 lb., although set at 250 lb. The valves upon being taken apart were found to be rusted tightly to their seats. This was undoubtedly due to the small amount of moisture left by the steam that was used by the maker to test the valve and set the spring, notwithstanding that the rust was dry when the valve was taken apart after failing to open.

The cast-iron valve disk with winged guides in the port would not stand the excessive chattering and hammering on its seat, to which most of the valves were subject when opening and closing, this being due to the surging of gas pressure that is always present in the discharge line of a compressor, especially at the higher speeds. One of the valves broke off its winged guide in the port and wedged the disk so that the line had to be shut off; examination showing that the metal of both seat and guide, which were of cast iron, were of good material and sound, the fracture being caused by the heavy chatter and pound as it opened and returned to its seat.

The capacities of all except the pop valve were low.

The pop valve gave practically no warning of when it was about to blow, and for this reason would be dangerous to anyone who might be in the path of the discharge when it opened. It was also shown that the seat of a pop valve could not stand the blow of the valve when closing, and remain tight.

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None of the valves, notwithstanding their low capacity, could discharge through a pipe line as tapped in their discharge outlet and maintain their setting and character. It was shown that a 2-in. valve would require a 6-in. pipe connection to carry its discharge gas the first 100 ft. with a proportionate increase for each added 100-ft. length, and this without taking into consideration bends or fittings, which would be necessary in most cases. Discharge pipe lines with test gages were erected so that this feature could be definitely demonstrated, the line being so constructed that readings could be taken for both 50-ft. and 100-ft. distances.

The regulations as written were based upon the knowledge gained from these tests, and it was found that to comply with them a different form of safety valve from any obtainable would have to be designed. The writer was selected to do this work, inasmuch as the valve then in use in our plants required only minor changes to enable it to meet the conditions as specified. This has been done, the valve, thoroughly tested, has been accepted and is now available.

The valve, as shown in Fig. 1, has no guide in the port, being guided in a bushing which is pressed in to the valve body, and by a cylindrical piston which is a part of what would be called the valve disk. This bushing is bored and reamed after being pressed in, and all alignment of valve is based upon this bushing, the seat being reamed through it as a guide, thereby making all parts of valve body and bonnet self-centering and self-aligning.

The valve is cone-pointed to present the least possible resistance to the flow of gas and is turned to an angle of 60 degrees, the seat being reamed $\frac{1}{32}$ in. wide and the angle being 57 degrees, allowing the valve to seat on a sharp edge and not on a flat surface. The piston guide is of sufficient excess area to get the characteristics called for in the regulations and to cause the valve to open wider as the gas discharge increases and the back-pressure builds up; the construction being such that the valve has these same features whether it is discharging through 500 ft. of pipe, or directly from the valve outlet.

Valve Acts Like Napier Orifice

It has the desirable feature for ammonia work, in that it opens gradually to any predetermined point as a lever valve, giving the operator sufficient warning to enable him to remedy the trouble in many cases; when this point is reached the valve continues to open as the pressure on the discharge side backs up on the increased area of the piston guide, the entire range being gradual and continuous until the back-pressure reaches 0.585 of the initial, giving a valve practically the capacity and character of a Napier orifice. The return movement is like the opening, and the valve leaves and returns to its seat without chatter or shock.

The noncorrosive metal was originated by the writer and is the same as has been in service in the steam and ammonia valves of the Quincy Market Cold Storage and Warehouse Co.'s plant for over eight years. It is not affected by any of the common corrosive elements and is hard—so hard that it requires grinding to get a good surface.

The capacity of this valve was demonstrated by tests in which the gas from two 26x18-in. single-acting cylinders at 60 r.p.m. was discharged through 100 ft. of 1½-in. pipe having seven elbows; the initial pressure being 272 lb. and the back-pressure on the discharge outlet of the valve being only 130 lb. per sq.in. gage, or in other words, the valve handled easily a displacement of 1,700 cu.ft. per min., with 24 lb. to spare on the outlet side of the valve, and this without chatter or shock or any other sign of distress.

The diffuser, Fig. 2, is a gas ejector designed to mix the issuing ammonia gas with a large proportion of air, so as to dilute and break it up thoroughly, and to do this with the least possible chance of obstruction by rain, snow, etc., and at the same time discharge upward. It is made of cast iron and in the various sizes called for in the regulations.

It will be seen that should the top become plugged by successive snow and rain, the gas could discharge

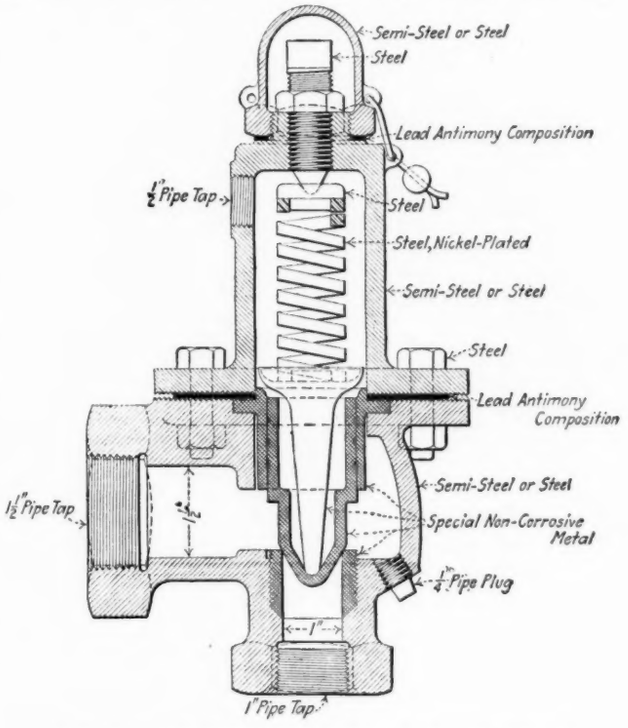


FIG. 1. THE QUINCY MARKET SAFETY VALVE

through the openings on the sides, issuing in jets or films and so mixing with the air. Each of these openings is large enough to pass the maximum discharge of the safety valve.

THE REGULATIONS AS FORMULATED

1. Definition of Ammonia Compressor—An ammonia compressor is any device which is capable of raising the pressure of ammonia gases by mechanical compression and which is designed, intended and used for such purpose. The term ammonia compressor, as applied to these rules, is intended to include both single- and double-acting types with one or more compression cylinders, whether they be of the reciprocating, rotary or centrifugal type, so called, or a combination of either or all, and will be considered to be that portion of the apparatus beginning at the gas-intake connection and extending as far as the first stop valve on the discharge pipe.

2. Definition of Safety Valve—A safety valve under these rules may be any mechanical device which will automatically prevent a pressure in excess of 275 lb. per sq.in. gage, which operates by allowing the escape to the atmosphere of the ammonia in excess of that pressure and which has been formally accepted by the Massachusetts Board of Boiler Rules.

3. Compressor Capacity—The capacity of an ammonia compressor shall be taken as the displacement per minute in cubic feet at the maximum speed at which it is operated.

4. Use of Compressor—When an ammonia compressor is used for any other purpose than ice-making or refrigeration, the conditions shall be referred to the Board of Boiler Rules.

5. Safety-Valve Location—(a) Every ammonia compressor shall have one or more safety valves located on the discharge side of the compressor and between the displacing element or elements and the first stop valve, and be of such size and adjustment that the pressure cannot exceed 275 lb. per sq.in. gage.

(b) There shall be no stop valves of any description between the safety valve or valves and the displacing element, or elements.

6. Materials and Design—Safety valves shall be constructed of materials which are not likely to be corroded by either the gases or liquids with which they come in contact and shall be designed with an ample factor of safety for the pressure, vibration and operation to which they may be subjected.

7. Sizes—Safety valves which discharge to the atmosphere shall be not smaller than ½ in. nor larger than 2 in. nominal diameter, and shall be of the direct spring-loaded type, but not what is known as a "pop" valve.

8. Capacity—If more than one safety valve is used, their combined capacity shall be taken as the discharge capacity required by paragraph 21 of these rules.

9. Inlet Connection—(a) Every safety valve shall have a full-size direct ammonia inlet connection at least equal in area to the nominal inlet of said valve.

(b) When two or more safety valves are used on a common inlet connection, this connection shall have a cross-sectional area equal to or greater than the combined area of the nominal inlets of all said valves.

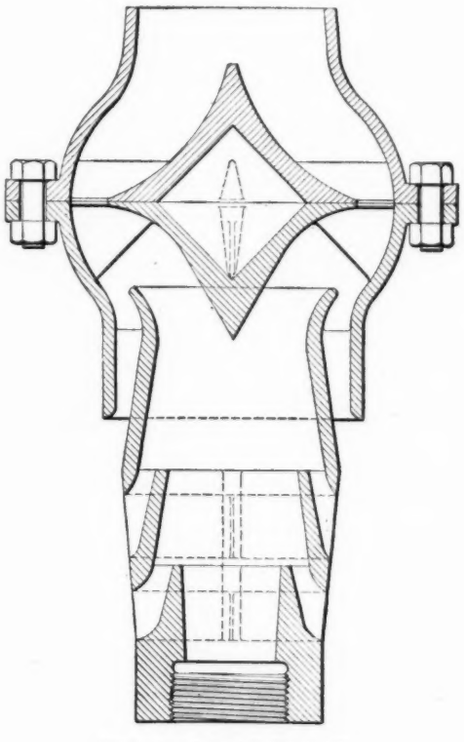


FIG. 2. THE DIFFUSER

10. Safety valves shall be installed with as short an inlet connection as practicable, to avoid serious vibration stresses; extra heavy pipe and fittings shall be used.

11. Opening Pressure—Safety valves must begin to open at a gage pressure not to exceed 250 lb. per sq.in.; and the lift of the valve disk from its seat at 275 lb. per sq.in. gage-pressure shall be at least ¼ of the nominal diameter of the valve.

12. Tests—Each safety valve shall be tested and set by the maker in accordance with these rules, before shipment from the factory, and stamped "Massachusetts Standard Ammonia 250-275 lb.," together with the maker's name; and when so tested, set and stamped, the bonnet shall be sealed, preferably with a wire and a lead seal bearing the maker's name and date of sealing.

13. Seal Not to Be Broken—After leaving the maker, the seal shall not be broken except for necessary testing and adjustment; and when so broken, the boiler inspection department of the district police shall be notified, so that the valve may be tested and properly sealed again with the seal of said department.

THE SAFETY VALVE DISCHARGE

14. Safety-Valve Discharge—(a) Every ammonia-compressor safety valve shall discharge the ammonia to a location and in such manner as not to endanger life or become a public nuisance. The discharge pipe shall be carried at least 10 ft. above the highest opening in the roof of the building in which the ammonia compressor is located, and to an elevation at least 10 ft. higher than the roofs of adjacent buildings, provided such adjacent buildings have an opening or openings in the walls, through which ammonia may enter the interior of the said buildings, excepting as hereinafter provided.

(b) A plant having not more than 25 lb. of ammonia in its working system, may have the safety valves discharge into an areaway unrestricted and open at the top, or into an open space between walls, provided such opening or areaway is at least 100 sq.ft. in area, and that the discharge pipe is located at least 6 ft. from the nearest wall having an opening into the interior, also that the discharge end of said pipe is turned upward.

(c) A plant having more than 25 lb. of ammonia in its working system may have its ammonia safety valve or valves discharge as herein provided, if such opening or areaway and such discharge-pipe outlet are located, from the nearest wall having an opening into the interior, not less than as many times 100 ft. and as many times 6 ft., respectively, as 25 lb. are contained in the actual working charge; it being intended in all cases to have an open and unrestricted vertical column of air above the level of the ammonia-discharge outlet, the horizontal area of which column shall be not less than 100 sq.ft. multiplied by the number of times 25 lb. are contained in the weight of ammonia in the working system, and of a horizontal radius not less than 6 ft. multiplied by the same factor.

(d) The ammonia gases may discharge into a ventilating stack, the inside diameter of which is at least 12 times the diameter of the ammonia-discharge outlet; the first possible outlet of said stack, above the discharge end of the ammonia-pipe being not less than 10 ft. above the roof, or as before provided in this paragraph; and when such stack is used for this purpose, the discharge end of the ammonia pipe shall be turned upward.

(e) An ammonia discharge pipe shall have no connection whereby the gases of discharge may be subject to ignition. No connection shall be made to any pipe which is a part of a steam or plumbing system or to a pipe or stack which is likely to be seriously corroded or obstructed.

15. Diffuser—(a) When ammonia is discharged into the atmosphere through a pipe other than a ventilating stack, a diffuser shall be used, which shall be attached to the upper end of the discharge pipe. This diffuser shall be of such design and construction that there may be no restriction of the flow of gases, the purpose of this diffuser being to intermix the ammonia gas with air and prevent the discharge of an undiluted column of ammonia gas.

(b) The design of this diffuser shall be such that the diameter of the ammonia nozzle shall be approximately the same as the nominal diameter of the safety-valve outlet.

(c) When the diffuser is used as the outlet for more than one safety valve, the area of the ammonia nozzle shall be approximately that of the combined area of said valves, based upon the nominal diameter of their outlet.

(d) The combined area of the air inlets shall be approximately double the area of the ammonia nozzle; and the total discharge area shall equal the combined areas of the air inlets and the ammonia nozzle.

(e) The diffuser shall be attached in a vertical position and be of such design and construction as to prevent obstruction of its openings by rain, snow, ice or otherwise; and it shall be constructed of material which is not likely to be seriously corroded by either the gases or liquids with which it comes in contact. It must also be accepted by the Massachusetts Board of Boiler Rules, be stamped "Massachusetts Standard" and with the maker's name.

(f) The type of diffuser which will be satisfactory to the said Board is one embodying the principles shown in Fig. 2.

16. Attendance and Means of Escape—If the ammonia compressor, while in operation, is under the supervision of an operator in attendance at all times, the safety valves may discharge directly into the compressor room, provided there shall be no persons in the building containing said ammonia compressor, except those directly connected with the operation of said compressor plant, and provided also that there are ample means of exit for persons who may be in the building in which said ammonia compressor is located.

17. Tapping—Safety valves must have a ¼-in. pipe-tapping on the discharge side of the valve disk, which tapping shall be closed with a pipe plug, except when the opening is used for the purpose of testing the valve.

SAFETY-VALVE DISCHARGE PIPE

18. Connection—The discharge pipe connection to the safety valve shall be made as flexible as practicable, be connected with the least number of fittings, and in such manner as to prevent damage to the safety valve or its connection by vibration or other stresses likely to be transmitted.

19. Metals—The safety-valve discharge connections shall be of metal or metals not likely to be seriously corroded by the gases or liquids with which they come in contact, iron or steel being acceptable for this purpose.

20. Sizes and Lengths—(a) The area of the discharge pipe from a safety valve shall be not less than the area of the nominal discharge outlet of said valve in accordance with the table given herein; not less than this size of pipe shall be continued for the first 100 ft. of pipe or fraction thereof, and for lengths of over 100 ft., the pipe shall increase one pipe size for each 100 ft. or fraction thereof.

(b) When more than one safety valve discharges into a common discharge pipe, the area of said common pipe shall be at least equal to the sum of the areas of the individual discharge pipes of said valves; not less than this area of the common discharge pipe shall be continued to a point 100 ft. in pipe length or fraction thereof from the farthest safety valve, after which point the pipe shall be increased one pipe size for each 100 ft. additional length or fraction thereof.

(c) It is intended that there shall be no reduction or restriction of area of the discharge pipe or pipes, as here given, except as provided in paragraph 15, and that no elbow or tee shall be allowed in any discharge-pipe line.

(d) Where a change of direction is necessary a bend equal in radius to at least 5 pipe diameters shall be used; and when more than one safety valve is connected to a common discharge pipe the connecting fitting shall be a Y, but not a T.

SIZES OF VALVES FOR COMPRESSOR DISPLACEMENTS

(e) Safety valves must have their discharge outlets tapped or flanged for pipe connections as follows:

Valve Diameter Inches	Outlet Diameter Inches
½	1
¾	1¼
1	1½
1¼	2
1½	2½
2	3

(f) The type of safety valve which will be satisfactory to the Massachusetts Board of Boiler Rules is one embodying the principles shown in Fig. 1. In this valve, the pressure in the casing above the valve disk shall not be allowed to build up a pressure exceeding 5 lb. gage to the square inch on the back of the valve disk, this space being vented to the atmosphere through a suitable connection in accordance with the provisions of paragraph 14.

21. Minimum Sizes—Minimum sizes of safety valves to be used on any ammonia compressor, based upon cubic-feet displacement per minute, shall be as follows:

Size of Safety Valve Nominal Diameter Inches	Allowable Displacement Cu. ft. per Min.
½	120
¾	250
1	510
1¼	830
1½	1,200
2	2,120

22. Bypass Valve—(a) An automatic bypass valve may be used, adjusted to operate at any gage pressure under 250 lb. per sq.in.; but this valve shall not be accepted as the safety valve specified in paragraph 5.

(b) This bypass valve, when used, should conform in design, material and construction to the specifications given in paragraph 6.

The following emergency instructions have been prepared by the writer and appended to the Massachusetts Ammonia Regulations:

EMERGENCY INSTRUCTIONS

Ammonia is colorless as a gas, has a pungent and penetrating odor. Its specific gravity is 0.589, or about one-half as heavy as air, and therefore it has a tendency to rise. Advantage may be taken of this property, and wherever ammonia is escaping into still air the better air will usually be found at the lower levels. This feature should not be depended upon, however, for the reason that ammonia gas mixes with moving air readily and in proportion to the humidity, the moisture in the air absorbing the ammonia gas and carrying it to all points affected by the air movement.

If a helmet is not at hand, a thick towel saturated with vinegar and fastened over the mouth and nose will enable the operator to remain in the gas until the acid of the vinegar has been neutralized by the alkali of the ammonia, the time depending upon the amount of vinegar the towel will hold, and is generally from two to five minutes. A gallon of strong vinegar should be kept on hand for this purpose.

Ammonia causes irritation and congestion of the lungs and bronchial tubes, violent coughing, vomiting, etc., and, when breathed in sufficient quantity, suffocation and death. In case of partial or total suffocation by ammonia gas, remove the victim from the gaseous atmosphere at once, carrying him quickly to fresh, moving air if possible, and immediately give manual artificial respiration. Do not stop to loosen clothing, or otherwise lose time, as every moment of delay is serious. Call the nearest doctor immediately.

If approved mechanical devices for artificial respiration are at hand, they should be used; but the manual method should be applied at once, until the mechanical device is ready to apply or the doctor arrives to take charge of the case.

As soon as possible after starting artificial respiration, the clothing should be loosened, as well as any other restriction to the breathing or circulation of the victim. Oxygen should be given as soon as possible, to displace the ammonia gas in the lungs, the use of the oxygen to be discontinued as soon as the patient shall have revived and received plenty of fresh air. Artificial respiration should be kept up for a long time, if necessary, cases of resuscitation being on record where favorable results were not obtained until after two hours had passed. If natural breathing stops after being restored, use artificial respiration again.

Do not give the patient any liquids by mouth until he is fully conscious, after which he should be given mild stimulants and mildly acid drinks (vinegar and water being excellent) to neutralize the ammonia absorbed by his system. Give him at all times plenty of fresh air and keep the body warm.

The handling of liquid anhydrous ammonia is attended with very much greater danger than the gas, for the reason that a portion of the liquid instantly becomes strong gas which is constantly increasing, while the liquid itself, if it touches the flesh, produces a condition, by freezing, which is much the same in effect as severe burns caused by sulphuric acid or scalding water. This is particularly true where the liquid reaches the flesh through the clothing, as the clothing acts like a reservoir, much the same as with scalding water. For these reasons, a leak of the liquid should never be approached unless the body is protected by clothing enough to prevent the low temperature of the evaporating liquid from reaching the flesh; there should also be over the clothing a liquidproof covering which will prevent the liquid ammonia from soaking through. Ordinary woolen clothing covered by a rubber coat and rubber boots will answer for the average small leaks.

In addition to this, the face, eyes and lungs must be protected against both liquid and gas, preferably by an approved liquid and gasproof helmet, which will provide air or oxygen for breathing for a sufficient length of time to enable the operator to get to the trouble, apply the remedy and get back to a place of safety.

One of the safest methods is to cover the leak with as large a stream of cold (not hot) water as possible, until the proper valve can be shut and the leak stopped. This is also true of a gas leak if it can be located and cannot be handled otherwise. A fire hose with the stream of water covering the leak will absorb the ammonia readily, and is one of the best methods of handling a serious leak when other means fail.

Lubrication by Forced Circulation is the best method known, where the oil is delivered into the bearings under pressure in a copious stream, collected and returned after being filtered and cooled. Pressure and large passages insure an ample supply, and the comparatively cool oil conveys heat from the bearing in proportion to its weight, specific heat and temperature rise.

Wick or Capillary Oilers—Where the upper end of the wick is in contact with the shaft or moving part and the lower end is immersed in the oil which feeds upward, the bearing is virtually self-oiling on the principle of capillary attraction—similar to an oil lamp. Another form conveys the oil by a wick by the same principle from a reservoir to the elevated end of a tube into which the wick is thrust thence by gravity to the bearing in drops similar to the drop feed oiler.

Owego Light and Power Plant

By Gilbert Newell

SYNOPSIS—A remodeled combination hydro-electric, engine- and turbine-driven steam plant. Under the present arrangement the various generating units can be operated in parallel or singly. The engine belt-driven generator is used as a motor to drive an arc-light machine when water power is not available.

The plant of the Owego Light & Power Co., at Owego, N. Y., being a combination of water- and steam-driven units, the installations and operations are somewhat different from those in most plants. Fig. 3 shows two views of the turbine room after extensive alterations.

A number of changes from smaller to larger units have been made during the past five years to meet the demand for more power as the business increased. These replacements were expensive and have never been satisfactory.

About two years ago it was decided to discard the old-style 125-cycle, single-phase generators used for lighting, and the 500-volt, direct-current generator used for power, and to install modern uptodate equipment. Fig. 1 shows

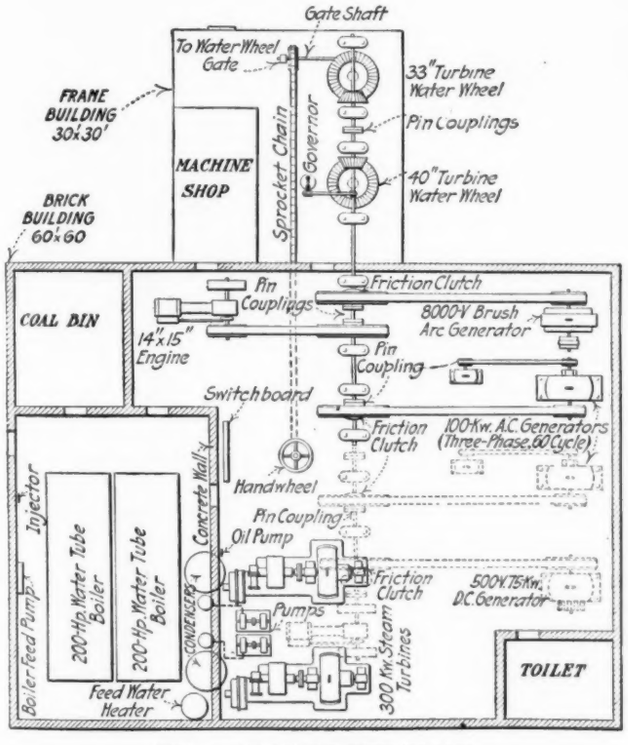


FIG. 1. PLAN OF THE PLANT

Heavy lines indicate present and dotted lines former arrangement

a plan of the plant before and after the last change. The dotted lines indicate the apparatus that was supplanted.

A line shaft ran above a 33-in., 150-hp. and a 40-in., 75-hp. turbine waterwheel, both of which were in a room adjoining the generator room. The generators were belt-driven by friction clutches and pin couplings. These units consisted of an 8500-volt, 5.5 amp. arc generator for street lighting; two 100-kw., 125-cycle, 1100-volt, single-phase generators for lighting; a 500-volt, 75-kw., direct-current generator for power; a 14x15-in. and a 15x16-in. engine with belt drive to the shaft. In the boiler room were two hand-fired, return-tubular boilers, operated at a steam pressure of 90 lb.

The water power fluctuates at different seasons of the year, and it was, and still is, the practice to use all that is available and make up the deficiency with the steam engines. The generator-driving pulleys on the main shaft

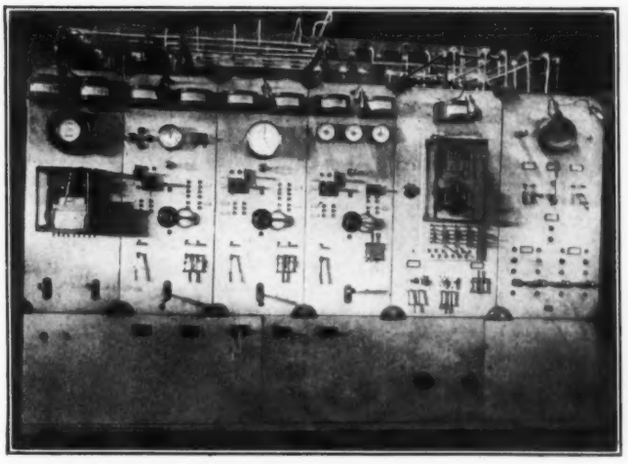


FIG. 2. SHOWING THE SWITCHBOARD

are equipped with a friction clutch and a pin coupling, the former being used to bring the generators up to speed, when the latter is brought into action, doing away with clutch-slipping. Any generator can be run independently of the others. A pin coupling between the two alternating-current generators could, before the change, be disengaged, allowing one part of the shaft to be shut down for necessary repairs, while the other could be run with the 15x16-in. engine, the waterwheels, or by the 14x15-in. engine, as the case might be.

It required considerable ingenuity to plan the change from the old lighting and power service to the 60-cycle, three-phase service for both, and from the low- to high-pressure steam, without seriously interrupting the service.

Two 200-hp. water-tube boilers with 180-lb. working pressure were installed, one at a time, in place of the old fire-tube boilers, and temporary pipes were run to the two engines and to one 900-kw., three-phase, 60-cycle turbine-generator set. The change was made Sunday morning, when there was the least demand for light and power. The line force made the necessary outside changes, while the engineering force installed a 100-kw., three-phase, 60-cycle, 2300-volt belted generator in place of one of the old single-phase units, and after a five-hour shutdown the new unit was put in operation and the 15x16-in. engine and part of the lineshaft were removed (see dotted lines Fig. 1), making room for a second 900-kw., three-phase, 60-cycle turbine-generator set which was installed. The turbines are each equipped with a multi-jet condenser operating with a 25- to 27-in. vacuum.

Oil is supplied to the turbines by a home-made gravity system. After it leaves the turbine it is caught in a tank,

from which it is pumped by a small duplex steam pump into a filter suspended on the wall, and after filtering it flows to a storage tank and is piped back to the turbines.

The switchboard, Fig. 2, is made up of six panels and is equipped with a synchronoscope, etc., so that the two

When no water power is available, turbines or the engine is used. When running the turbine the belted generator is used as a motor to drive the arc generator when the street lights are on, thus doing away with running the engine and turbine at the same time. Plans are under

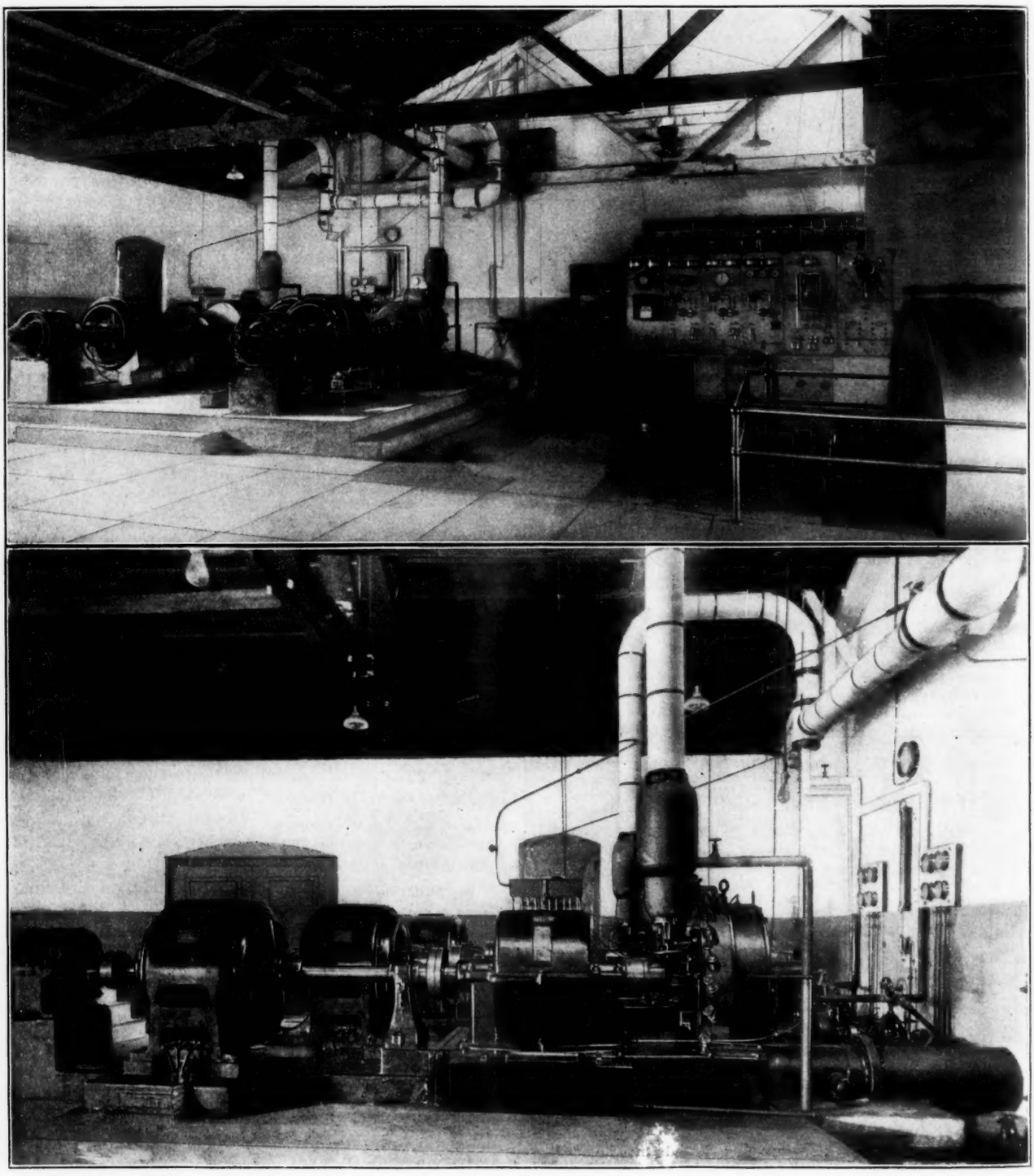


FIG. 3. TWO VIEWS OF ENGINE ROOM, SHOWING TURBINES AND END OF WATERWHEEL-DRIVEN SHAFT

turbines, or one turbine and the belted generator, or all three, can be run in parallel. The practice now is to use all of the water power available by running the belted generator, then help out with steam by running the 14x15-in. engine connected to the same shaft by a pin coupling, when practicable, or by running one of the turbines in parallel with the belted generator.

way to install an alternating-current outfit for street lighting, and do away with the arc generator.

The boilers are equipped with damper regulators and are hand-fired. Water is taken from a nearby creek by a 4x5-in. automatically regulated duplex steam pump, and is pumped through a feed-water heater in which the temperature is raised to 190 deg. F.

Editorials

Doing Things Electrically

"Electrical Prosperity Week," the time chosen to boost things electrical, is at hand. The nation-wide campaign, which has been pushed vigorously by the Society for Electrical Development for several months past, has brought forth enthusiastic response, and most of the large cities, as well as a number of the smaller towns, are coöperating, with varied programs. These include electrical exhibits, special illuminations, window displays, moving pictures and wiring campaigns—in short, anything that will impress upon the public the desirability of doing things electrically.

The movement is essentially commercial, affecting directly the manufacturer of electrical appliances, the contractor and the central station, but indirectly it should benefit the industry as a whole. Although one is accustomed to look upon the use of electricity as almost universal, the number of instances, even in metropolitan districts, where it is not used for domestic lighting, is surprising. Moreover, the field for domestic uses is almost unlimited. Provided the price can be made sufficiently low to compete with gas and coal, greater use of electricity in these lines will tend to increase the use factor of the small-consumer group of customers, and should automatically tend to lower their rates.

Interstation Inspection Visits

Would that employers of operating engineers could be made to realize better the benefits of interstation inspection trips. How often do some engineers get away from their plants for a half-day to watch the work of others in their vicinity, to discuss points of common interest right on the ground with other engineers and to bring home hints for the betterment of their own properties? Not often enough, it is safe to say.

The confining nature of the engineer's work is one of its greatest drawbacks, but in these days of telephones and automobiles an engineer can safely be allowed now and then to visit plants within quite a wide radius of home without serious danger of trouble. Without for a moment decrying the excellent work of local organizations of engineers meeting after hours in the evening for professional interchange and the spread of experience and the development of human intercourse, it is important that every technical worker with executive responsibilities in this field should be given reasonable opportunities to see for himself, on his employer's time, what other installations are doing. There is no reason why this work should be done on the engineer's own time during his rare absences from his home station during business hours. The employer should recognize the wisdom of such visits and include them in the time for which he compensates his engineer.

The central station that sent its chief engineer recently to an efficiency convention fifty miles from home did well. The convention devoted probably not over thirty minutes to the discussion of power-plant problems, but during that time a lot of first-hand discussion came up relative to combustion problems and the engineer took part in it, besides listening to the views of others. He was away from the plant probably half a day for that thirty minutes, plus such opportunities as were at hand to see other plants and to talk things over informally in the corridor of the hotel; but it was worth while. It was not a stationary engineers' convention, but it was worth sending the chief to, even for so short a time, and other plants might well go and do likewise. A capable engineer deserves more than to be buried in his station from one month's end to another, and it is gratifying to seen signs of improvement in this respect.

Especially should interstation visiting be encouraged when changes in plant are in contemplation. Generally speaking, it is a great mistake to rush ahead into purchases of equipment of a new or different design from the previous installation without availing oneself of the opportunities, so far as they exist, of seeing similar apparatus at work in the reasonably near neighborhood. Let the operating chief, as well as the consulting engineer, visit such installations, report upon them in his own words, and thus contribute to the engineering solution of the local-plant problem. Fifty dollars expended in traveling expenses to obtain inside operating information through an experienced engineer who is to handle the machinery after installation is a small price to pay for the possible good to be derived from a liberal visiting policy.

The New Anthracite Sizes

According to *Coal Age* there is a movement on foot to revise the present standard sizes of anthracite by reducing the number approximately 50 per cent. This would interest all power plants, especially as two of the proposed alterations affect sizes that may be considered power-house coals. It is proposed to combine chestnut and pea in one size and to mix buckwheat Nos. 2 and 3 to form "steam coal." Other suggested changes in the larger sizes are not of such moment to the power house.

To study the effect on the cost of producing power, it may be assumed that the prices of the new sizes will be the average of the present prices of the coals it is intended to mix. Even disregarding the extensive alterations to existing grates that would be necessary to burn the new sizes successfully, the changes proposed will hardly be welcomed by power-house operators, and they are important consumers whom the mine operators cannot afford to ignore.

The many power and heating plants still using pea notwithstanding its decreasing popularity as an economical power-house fuel would be particularly severely handicapped under the proposed standard. Those paying the New York price would spend nearly twenty-five per

cent. more. The one redeeming feature of the plan is that plants now using pea would probably adopt buckwheat No. 1 when they experienced any difficulty in obtaining pea. Such a change would ordinarily promote greater efficiency and economy, but this is not the object of the proposed revision of anthracite sizes.

Buckwheat Nos. 2 and 3 mixed makes an excellent steam coal even now used extensively and could profitably be used more generally, but withdrawing buckwheat No. 3 from the market would materially increase the fuel bill of many stations for it is probably more commonly burned by power or heating plants than the larger No. 2. Changing over No. 3 to the new "steam coal" would add about ten per cent. to the coal bill at New York prices. Where buckwheat is now successfully burned this would be a dead loss, for the new fuel would be no more efficacious. In some plants in which straight buckwheat No. 2 is now burned, the new size would tend toward economy, but not enough to offset the sacrifice made by the greater number of plants that would have to abandon using buckwheat No. 3, nor as much as would be realized should straight buckwheat No. 3 be made the plant fuel.

This scheme of the coal operators looks much like an attempt to swell their profits at the expense of their best customers and the most important consumers of anthracite, by compelling them to use a more expensive coal. Such an attempt can only invite stubborn resistance, for the power-house interests will certainly have some well-grounded objections to raise.

"Knocking" One's Profession

Were the engineering profession not a good one, it could not survive the "knocks" given by many in it when asked for their candid opinions of its opportunities. It is human nature to think that what might have been would have been better; experience brings many disillusions in every walk of life, and so, many men advise their sons to steer clear of their occupations. Engineers do this as well as doctors, lawyers and other specialists, and sometimes to the lasting loss of their own great industry. Why is this so?

There never was a time when ability was more in demand in the engineering world than today, nor when the outlook for fair compensation was better. Here is Smith, in charge of a five-thousand-kilowatt power plant, holding the confidence of his employers, given a pretty free hand in administration of mechanical and electrical matters and earning enough to give his children a good start in the higher educational fields, besides carrying something to the savings bank now and then. His son has a natural bent for mechanics, has in a sense been brought up on a partial diet of steam engineering and electricity and goes to the "old man" for advice about his future work in the world. What does Smith tell him?

Well, one kind of Smith tells the boy that there is mighty little in power-plant engineering except hard work, exacting hours, labor difficulties, dirty hands and clothes, poor pay, continual struggles to keep valves and joints tight, constant changes in the demands of efficient combustion, shifts and turns of fortune, with the central station always on the trail like a hungry wolf, waiting for the breakdown which will insure its capture of the whole outfit, after which—exit the engineer, unappreciated and soon forgotten. Ten to one, this kind of Smith then paints a glowing picture of the rich profits of the law, describing how young Blackstone represented the company at a commission hearing and defeated the local central station in its efforts to prevent the construction of an underground conduit line across a street between two of the factory buildings. He sets forth the alluring fact that after the hearing Blackstone got a raise in salary, while he, Smith, who prepared all the engineering data that Blackstone used, seemed to have only his work for his pains. And so on.

The other kind of Smith looks deeper. He sees that his boy has no more talent for legal affairs than Smith himself has for writing successful plays, but he knows that his son is a *born* mechanic; that he can't keep his hands off tools, wood and iron, and that mathematics comes as easily to him as French does to his sister. Smith has an imagination that works to some purpose. He tells the boy that it will be many a long day yet before the central station can shut down all the big or all the little power installations, in view of the ever-present heating problem; that more and more power is being used by mankind in its daily work; and that the world is looking for the man who can show it how to produce the unit of output a fraction cheaper than before, taking all the facts into consideration. With proper education, young Smith can take these facts into hand better than his less adapted brother, perhaps with an originality lacking in his father.

Will Smith, Senior, spoil a good engineer by dwelling on the obstacles mere hard work puts in his path? Not for a moment. Such a man knows that every occupation in life worthy of a man's efforts involves drudgery, disappointments, petty annoyances, and at times seemingly inadequate compensation, not to say downright poor pay. He also knows that, given a bent in the direction of his profession, supplemented by hard work and study, enthusiastic interest in opportunities small as well as large, with the broad-minded outlook that distinguishes the professional man from the mere wage seeker, plus determination and staying power, his son may rise to the highest plane of usefulness, and perhaps attain recognition throughout the industry.

More and more the engineer is taking an interest and playing an active part in public affairs; greater days are dawning, and never was there less excuse for throwing cold water on the interested inquiries of the younger generation relative to the opportunities of this important and honorable calling.

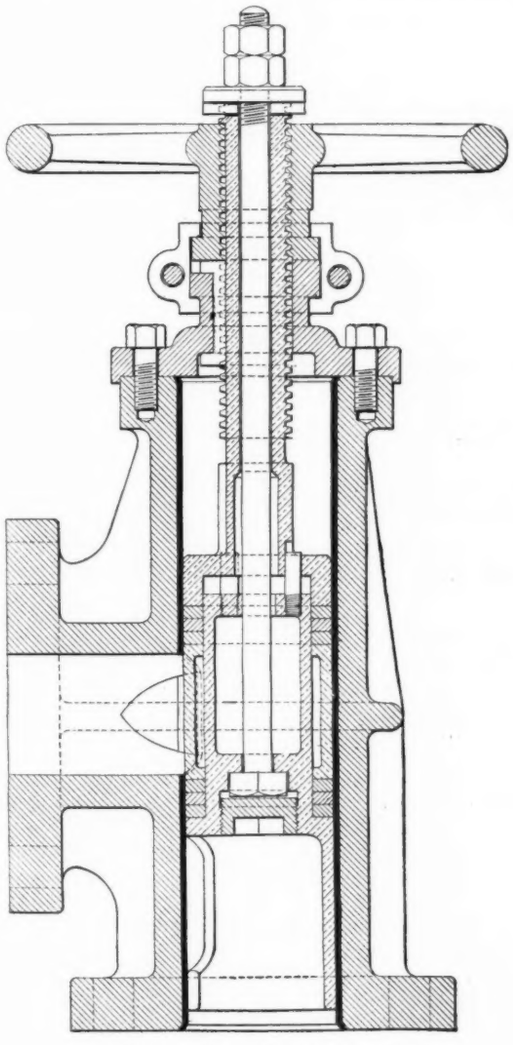
There are under construction or consideration two or three plants in which steam pressures of over three hundred pounds and two hundred degrees of superheat are to be used. As the ratio of the heat convertible into work to the heat required to produce steam under these conditions of higher pressure falls off rather rapidly as these high pressures are reached, it is likely that these figures approach the limit of successful development in this direction.

Some earlier sawmills used turbine waterwheels after the passing of the old paddle wheel, and now the modern sawmills are taking up with turbine engines and electric transmission.

Correspondence

What Broke the Valve?

Referring to the letter of N. Olson, which appeared at page 309, Aug. 31 issue, and the further discussion letters, page 451, Sept. 28 issue, from my knowledge of the type of the valve under consideration, if the piston packing rings, arranged as shown in the section herewith, were too tight, causing undue friction from tightening the nuts at the top of the rising stem, the effect of which cannot



SECTION OF BLOWOFF VALVE

be gaged, then when the valve was closed, excessive stress was brought on the collar or rabbet of the threaded handwheel.

As the collar was of cast iron only ⅜ in. thick, the blowoff valve of No. 2 boiler was probably fractured before No. 1 blowoff was opened. When the bottom of the piston in the closed position received a shock or pressure from opening the other blowoff valve, the piston was forced upward, carrying with it the handwheel on the screwed stem, for there was nothing to hold it in position. I do not consider the arrangement of the blowoff pipe and connections had anything to do with the breaking of the valve. W. J. Jones.

Philadelphia, Penn.

In answer to the question, "What Broke the Valve?" I would attribute the cause to pipe layout. In the first place, the blowoff line is much larger than necessary, 3-in. pipe being large enough for the main line. A water-hammer will be created in the short section to No. 2 boiler whenever No. 1 boiler is blowing. This same condition is likely to happen in No. 1 and No. 2 boilers whenever a blowoff valve is opened farther down the line.

As a temporary means of overcoming this trouble without making any pipe changes, I would suggest placing a ⅜-in. vent in the blowoff line beneath No. 1 and No. 2 blowoff valves. These vents can be closed after a blowoff cock is opened, but where convenient they may be left open to a separate drain.

Pittsburgh, Penn. William A. Butler.

Cleaning Ammonia Condensers

The discussion by Mr. Anderson and Mr. Blanchard, in the Sept. 7 and Oct. 12 issues respectively, relating to the cleaning of double-pipe ammonia condensers shows the necessity of a few suggestions as to how best to do this job. I have never found it necessary to clean the ammonia space of any condenser with sand, steam or air. The oil that collects in the condenser does not amount to much, but it can and should be removed before the winter's overhauling begins.

Heat the discharge of the compressor to 250 deg., and then, with the machine running slowly, shut the cooling water off one condenser stand at a time. If the coil does not heat throughout, shut the ammonia outlet valve on one or two of the other stands, which will allow ammonia to store and give a more rapid circulation to the hot gas. Any oil in the condenser will be heated and circulated with the ammonia and finally lodge in the evaporating coils or in the oil trap.

It is customary to steam out the evaporating coils once a year or whenever necessary to take out the oil. Do not try to steam out condenser coils. The main objection to steaming out double-pipe condensers is the injury done to the gaskets or packing in the stuffing-boxes where the inner pipe passes through the ammonia return bend.

Sand or any like substance has no place in the ammonia system and should never be used.

The inner surface of the single-pipe ammonia condenser is cleaned of oil in the same way as the double-pipe. Usually little hard scale will collect on the water pipes of the double-pipe condenser. A slush will accumulate, but that is easily removed by changing the flow of water and washing the coils out once a week.

If hard scale does form, the use of a turbine tube cleaner is advisable. A twist drill, secured on a length of ½-in. pipe and operated by an air or electric motor, will remove hard scale quickly and thoroughly. Sand may remove this scale, but it is not certain, and if a turbine cleaner or drill is used one knows what the results will be. Removing the water-return bends on a double-pipe condenser is not a hard job. Graphite and oil should be used on them.

Scale that forms on the outside of single-pipe condensers can generally be cracked off by using the coils as an evaporator for a short time. The unequal expansion and contraction of the scale and the pipe will loosen the former, and a little tapping with a light hammer will knock it off.

The use of a reliable brand of condenser paint will be found excellent to make condenser cleaning easy.

Chicago, Ill. A. G. SOLOMON.

Quick and Cheap Repair Job

In the issue of Oct. 26, page 594, R. L. Ellis tells of a quick and cheap repair on a Heine boiler, but he does not tell how the tubes were cut out and expanded. I suppose it was done the same as we do usual repair work; that is, by hand.

If that is the case, I should like to ask men who do boiler tubing if they ever expand one end of 138 tubes in 10 hr., or about four minutes per tube end. Of course a whole lot depends on the size of the tube; but for this type of boiler I don't think they were less than 3 in.

When I do tubing or other boilermaker's work I get boilermaker's wages, not $1.75 per day, but I never made so good a "record" as this and I can handle an expander some too. In a Stirling boiler with 3¼-in. tubes just installed, it took our men nearly ten minutes per tube end.

The reason I take this matter up is that some plant owners that see this record and repair bill will think their men are shoemakers and not boilermakers.

Berthierville, Canada. JOSEPH GISZTN.

The Unappreciative Superior

The editorial in *Power* of Nov. 2, under the caption, "The Unappreciative Superior," touches a subject of live interest to every engineer. Is there a man among us who has not at some time or other been unappreciated? As the editorial points out, anyone can get discouraged, but it takes will power to fight on all the harder when nothing pleases the man higher up.

A recent issue of *The Telephone Engineer* very aptly puts the matter this way: "If the world does not recognize your talents, don't get discouraged; get mad. An angry man sometimes accomplishes something; a discouraged one never does."

The dual relationship of employer and employee demands for efficient service a complete appreciation of the mutuality of interests. Service of the higher order is not entirely a matter of the server alone but calls for the full and sincere coöperation of the party served. Very many employers do not seem to realize this, and if they coöperate at all it is only at the point of the bayonet.

I had this particular phase of the matter thoroughly impressed on me while visiting a rejuvenated power plant operated by a sincere although somewhat strenuous engineer. Finding the plant quite uptodate and recalling its sad condition a year ago, I inquired of my friend how he had induced the management to buy modern equipment. The reply, stripped of that verbiage peculiar to those habitually frank, was:

"Well, I fought for 'em, every mother's son of 'em. I talked myself blue in the face and hoarse as a frog and quoted high-economy records and B.t.u. up the chimney and dollars to be saved, etc., etc.; and then the boss said no, he could not afford it because the power plant was an expensive proposition anyhow and to put in high-falutin' machinery would be like throwing money in the creek. So then I quit the job, and the boss said that if I felt that bad about it, he would buy a feed-water heater. So we got a feed-water heater.

"Then I wanted some materials to stop up the large cracks in the boiler settings, but he couldn't see it that way at all, because the air had to get into the boiler somehow and he could not see how it made any difference whether it went in through the draft doors or some other holes. So I quit again, and he hired me back with mortar and fireclay. And so it goes on and on—for every single improvement in the plant worth while I have had to quit my job in order to get it."

Such complete sincerity and self-sacrifice for what is right are quite rare, but such pitiful absence of coöperation is only too common. E. HURST.

Montour Falls, N. Y.

Exhaust vs. Live Steam Heating

In the issue of Oct. 26, I notice an article on page 590, "Does Live-Steam Heating Boost the Coal Bill?" and suggest that it be re-read, as it merits close attention. We have had some very good and practical experience here in our ten-story office building, in which we have our own plant and do all the work of the plant and considerable outside work. I will quote from our files two representative records. We keep close and complete records of all operations, such as weather, weight of fuel both day and night (we operate continuously) and weight of the ash; we meter the water and the steam output, use recording gages and keep CO_2 records, as well as many others. The comparisons that I am sending you are correct, and I personally know that the Sunday runs are comparatively harder and we earn less. I am not guessing at it, and if anyone doubts, let him come and take the scoop and fire the two runs and see.

Heating is done largely by the exhaust steam during the weekday run and largely by the direct steam reduced to ½ lb. (we use a vacuum system) during the Sunday run, and the weather was very near the same on each of the two days compared and the same fuel was used.

	Date	Lb. Coal Burned	Cost	Kilowatts Generated
(1)	Monday, Feb. 2	10,331	$16.87	1,070
	Sunday, Feb. 1	9,347	15.81	400
		984	$1.06	670
(2)	Monday, Feb. 9	13,876	$22.04	1,080
	Sunday, Feb. 8	10,745	17.46	330
		3,131	$4.58	750

In comparison No. 1, to do the same heating we used but 984 lb. more coal on Monday and generated 670 more kw.-hr. for a difference of $1.06. In No. 2 we used 3,131 lb. more coal on Monday and generated 750 more kw.-hr. at a difference of $4.58.

Why contend that current can be purchased and heating done direct for less cost than by a combined generating and heating plant? ASA P. HYDE.

Binghamton, N. Y.

It is rather confusing when one writer proves that exhaust steam is better than live steam for heating purposes, while another one proves live steam to be as good as exhaust and some contend live steam is best. A common-sense view of the matter is that exhaust steam will show advantages where the radiators are small, as the pulsations from reciprocating engines break up the high-resistance film that covers all heat-transferring surfaces. In boilers, surface condensers, refrigeration coils, etc., more B.t.u. will be transferred by the agitation of the liquids or gases on the surfaces of the metal. The higher the speed of the gases the more B.t.u. will be transferred per foot of heating surface. Also the better the circulation the more heat is extracted by the water from the conducting surfaces. Thus the breaking up of this resistance film plays a most important part in connection with heat-transferring surfaces.

A building designed to use live steam will therefore require a little larger radiators than if designed for exhaust steam or else must carry a higher pressure for the same results, and superheated steam would require still larger radiators. Slightly superheated steam should prove economical where it has to be conveyed long distances before being used, although requiring larger radiators, but at present it is not in favor.

In reference to the proper way to measure and compare the leakage of superheated and saturated steam referred to by C. E. Anderson in the issue of Sept. 14, p. 383, I believe it should be measured by the loss of heat units per minute represented by the water as shown by the steam tables. HARRY ALLEN.

Chicago, Ill.

The Business Side of Power Plant Engineering

I was very much interested in your editorial, "What the Modern Power-Plant Engineer Must Know," on p. 590 of the Oct. 26 issue. I had imagined that I was pretty well "up" on the business side of engineering, but some of those questions would put me guessing and if my getting the position depended on my answering them right off the reel, I believe I would lose out. For instance, there is such a wide latitude between plants of the same appraised value with regard to "the interest on the investment" expected. A few years ago I operated an electric plant where the stockholders expected a yearly return of about 350 per cent., while the plant I am operating now, although valued at $100,000, is only considered as a means of enabling the rest of the workings to produce dividends on the total investment, and has never been considered as an income producer for the stockholders. Then there is the question of how to arrive at the cost of refrigeration per cubic foot per year, and the proportion of fixed charges to be credited to depreciation. Some good consulting engineers allow 10 per cent., while others claim that 5 per cent. is enough and still others that 2½ per cent. is sufficient.

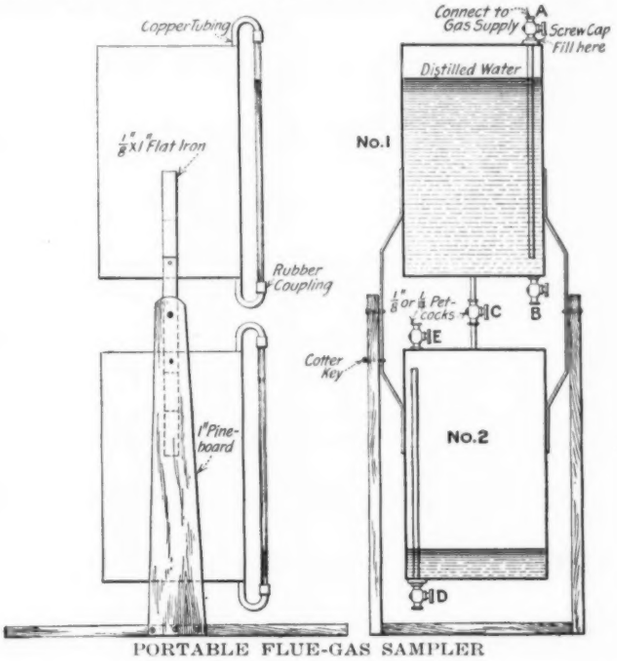
How can an engineer obtain this information so as to be able to give a concise and intelligent answer to each of the questions asked, and especially the ones named here?

Ringwood Manor, N. J. A. A. BLANCHARD.

Flue-Gas Collector

I notice in the Oct. 5 issue of *Power* a sketch and explanation of a flue-gas collector, by F. W. Fischer.

This apparatus would be better if a change was made in its design by continuing two tubes from the stop-cocks *A* and *D* to a point near the bottom of the collector in each position, as shown here. Otherwise the rate of flow of the water through the cock *C* and the resultant rate of aspiration of the gas sample into the container through the stop-cock *A* would vary with the head of water above



PORTABLE FLUE-GAS SAMPLER

cock *C*, and thereby obtain a sample of gas which would not be a true average through the entire period of aspiration.

With the improvement as shown, there will be a constant rate of aspiration with any setting of the stop-cock *C* so long as the level of the water does not fall lower than the bottom of the tube. C. H. BEAN.

Chicago, Ill.

Cutting Down the Cylinder Oil

In the Nov. 2 issue of *Power*, page 605, Mr. Anderson in his power-plant description writes about cylinder-oil consumption. Our plant uses more cylinder oil than it should, and if Mr. Anderson would tell more of the conditions in his plant, it would show whether those in ours are in any way similar.

Quoting Mr. Anderson's article, "A study was made of conditions and the conclusion arrived at was that too much cylinder oil was being used. Then the lubricator on the low-pressure side of the cross-compound engine was removed, . . . The saving also included the nonuse of oil cups above the piston rods, which were declared unnecessary as the rod packing contained enough lubrication."

How many drops was the high-pressure lubricator feeding before and after removing the low-pressure lubricator? How often is the piston-rod packing removed and replaced with new? Suppose the packing on the piston rod was metallic, how would it do to remove the piston-rod cups? The steam pressure is similar. Cutting down the cylinder oil on either high-pressure or low-pressure of our engines would cause the cylinder to groan and the valves to rattle. What can be done to remedy matters? G. T. MICHAEL.

New York City.

Testing Condenser Tubes

In the issue of Nov. 2, page 624, Junior Parrish tells about testing surface condenser tubes, but I think his way is rather expensive. I have tested surface condensers by filling the steam chamber with water, both heads being removed, and one is able to detect all the leaks in tubes and tube packing. If local conditions permit, a pressure of 5 or 10 lb. should be used, which will indicate the smallest leaks. WALTER WOLGAST.

Elmira, N. Y.

Instruction Cards Helpful

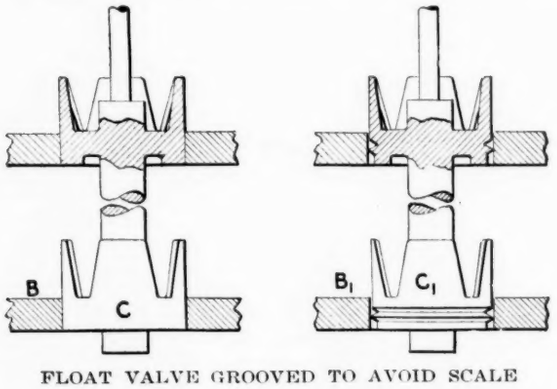
The suggestion by F. W. Fischer on page 418, Sept. 21, that manufacturers should furnish suitable instruction cards for use in connection with their equipment, applies especially to us out here on the Pacific Coast, for the reason that in most cases we are operating machinery a long distance from where it was manufactured, and are at a disadvantage when questions arise.

Full printed instructions, bound with tin or framed and fastened on the walls, would not be lost with the changing of help, but would be kept in the proper hands.

Portland, Ore. H. L. SHEPARD.

Scale Formed on Float Valve

A few days after installing a 20,000-lb. feed-water weigher to handle water at 210 deg. F., I noticed that the float valve, of the standard balanced type, was not working properly. The disks do not close against a seat, but fit closely inside of a chamber, as in a piston valve or the valve of a throttling governor. When the valve was removed the cause was readily apparent for scale had formed around it.



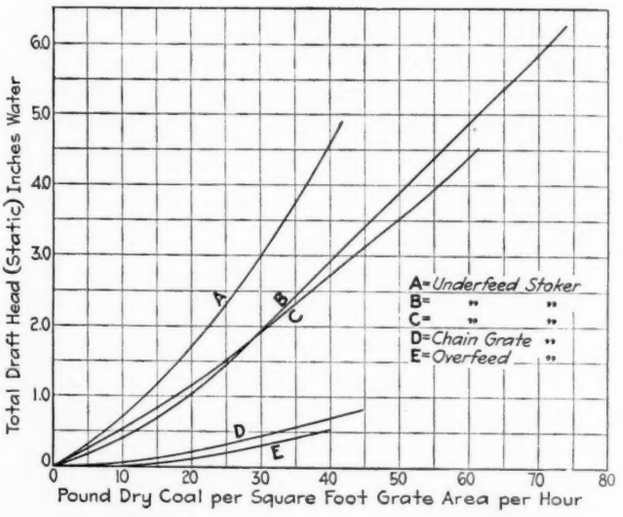
FLOAT VALVE GROOVED TO AVOID SCALE

A few days after cleaning this off the same trouble made itself manifest again, so I decided to try the expedient of cutting down the guides and the valve-disk area, making the disk proper a V-shape, which proved all that could be desired. About once a week we force the valve over the whole range of its travel from wide open to closed, which, on account of their shape, breaks the scale from the disks. CHARLES L. JUNO.

Milwaukee, Wis.

Limiting Factors in Forcing Boilers Above Rating

In reading my article entitled "Limiting Factors in Forcing Boilers Above Rating," in the Nov. 2 issue, I find that I have made an error in Fig. 4. Curve *A* should read, "B. & W. 600 hp.—21x14 tubes," and curve *B* should read, "B. & W. 520 hp.—21x12 tubes."



COMBUSTION RATES IN DIFFERENT STOKERS AT VARIOUS DRAFT PRESSURES

I also find that curves *D* and *E* in Fig. 1 are incorrect owing to an error in plotting. The corrected curves are here shown. THEODORE MAYNZ.

New York City.

Packing Trouble and Remedy

Considerable trouble is experienced in keeping the packing from leaking on outside-packed pumps, even when the packing glands are tightened until they are nearly (or actually) broken when the plungers are worn smaller in the middle, while the ends remain nearly their original size. Stiff springs under the gland nuts will relieve the strain at the two ends considerably.

The remedy is to turn the bad plungers down until the diameter is the same throughout their entire length. After this is done it will be found that the pressure and friction required will be reduced from 30 to 60 per cent. If considerable material has to be removed from the plungers, it may be necessary to bush the glands and bottom of the stuffing-boxes or have a brass, babbitt or iron washer made a close fit on the plunger and the inside of the stuffing-box and about ⅛ in. thick. This will not only save trouble, but will reduce the packing bill. JAMES E. NOBLE.

Toronto, Ont.

Inquiries of General Interest

Removing Stains from Orsat Burette—How can the dark smoky deposit be removed from an Orsat burette that has been used for flue-gas analysis? J. C. R.

Oily deposit can be removed with gasoline. The smoky appearance due to a deposit of carbon can be removed with kerosene. If the glass has become discolored while hot, then for removal of the stain it may be necessary to soak the burette in the cleansing liquid for some time, occasionally scrubbing the discolored surfaces with clean cotton waste.

Back-Pressure Valve Combined With Reducing Valve—What objection would there be to having an exhaust back-pressure relief valve set at 6½ lb. pressure and a high-pressure reducing valve reducing live steam to 8 lb. pressure, both connected with the same low-pressure heating main? J. J. M.

The back-pressure valve would open when the pressure was in excess of 6½ lb. per sq.in., and whenever the pressure in the heating main was higher, live steam delivered through the reducing valve would increase the back-pressure of the engine and would escape through the back-pressure valve to the atmosphere.

Use of Compressed Air in an Injector—Will an injector work when supplied with air under high pressure in place of steam? W. H. L.

Kneass, in his work entitled "Practice and Theory of the Injector," defines an injector as "an apparatus in which a gaseous jet impinges and is condensed by a fluid mass whose final kinetic energy exceeds that of a jet of similar form and density discharging under the initial pressure of the motive jet." Air is not condensed by water, and therefore the conditions required for operation of an injector could not be fulfilled by a jet of air impinging upon water, and to comply with the requisites the fluid mass would have to be liquid air.

To Determine Gallons Pumped per Pound of Coal—Knowing the size and speed of a plunger pump and the number of pounds of coal used per hour, what would be the method of computing the number of gallons pumped per pound of coal? D. T.

Multiply the cross-sectional area of the pump plunger in square inches by the stroke in inches and by the number of strokes per hour, and this product multiplied by 1 minus the percentage of slippage will be the actual number of cubic inches of water pumped. Divide by 231 (the number of cubic inches in one gallon) and the quotient will be the number of gallons pumped per hour. Divide by the number of pounds of coal used per hour, and the last quotient will be the gallons pumped per pound of coal.

Advantage of Steam Turbines Using High Vacuum—Why is not a high vacuum as advantageous in the operation of a condensing reciprocating engine as in the operation of a steam turbine? W. L. C.

In case of a reciprocating engine working condensing, without unduly curtailing the initial volume of steam admitted to the cylinder, the number of expansions is limited, and with a high vacuum the walls of the cylinder are alternately exposed to the temperature of the expanding steam and the cooling effect of vacuum, in consequence of which more than about 26 in. of vacuum counteracts the advantage of reduction of the back-pressure by higher vacuum. In the operation of a turbine there is a continuous flow of steam, and the higher the vacuum the greater the kinetic energy developed by the steam.

Selecting Size of Rivets—What controls the size of rivets to be selected for a riveted joint? R. S. H.

As there is a variation between the strength of the material of the rivets and the riveted edge of the sheet, no absolute rule can be given for selecting the size of rivets. To cover the practical considerations (1) the rivets must be of commercial size, that is, their diameter must measure in sixteenths of an inch; (2) the size of rivets must be within the capacity of the riveting machine as regards pressure required for properly setting up the rivets; (3) the pitch that results from the selected size of rivets must be small enough to afford a good calking pitch; (4) the size should be such as to secure maximum efficiency for the joints, that is, if possible, the rivet should be a crushing rivet.

Meaning of "Heavy" Steam Gage or Safety Valve—When a steam gage indicates more pounds per square inch than the actual boiler pressure, is it proper to say that it is "light" or "heavy," and how are these terms applied to safety valves? E. J. T.

In each case the terms are to be used according to the purpose of the appliance. When a steam gage indicates more pressure than it should, it is said to be "heavy," as a short method of imparting information that its indications are higher or "heavier" than the actual pressure. The operation or the adjustment of a safety valve is said to be "heavy" or "too heavy" if the pressure at which it blows is too high or too heavy for its purpose, or if the valve does not blow until the pressure of the boiler is greater than the pressure for which the valve is supposed to be loaded.

Measuring Insulation Resistance—When a voltmeter of 18,000 ohms resistance is connected across a certain circuit it indicates 115 volts, and when connected between one wire and the ground it indicates 10 volts. What is the insulation resistance of the other wire? H. C. M.

Letting V represent the voltage across the line (115 volts); v the voltmeter reading between one side and the ground (10 volts); r, the resistance of the voltmeter; and R the resistance sought, then

$$R = \frac{r\,(V - v)}{v}$$

Substituting,

$$R = \frac{18{,}000\,(115 - 10)}{10} = 189{,}000 \text{ ohms.}$$

Required Size of Chimney Flue—What should be the inside diameter of a round stack 81 ft. high, to be suitable for 100 hp.? R. G.

For a given height H in feet, the required effective area E in square feet is given by the formula

$$E = \frac{0.3 \text{ hp.}}{\sqrt{H}}$$

Substituting the values,

$$E = \frac{(0.3 \times 100)}{\sqrt{81}} = 3.33 \text{ sq.ft. or } 3.33 \times 144 = 479.52 \text{ sq.in.}$$

which corresponds to a diameter of about 24¾ in. As the actual diameter of a round chimney should be the diameter of the effective area plus 4 in., the actual diameter should be 24¾ + 4 = 28¾ in.

Greater Economy Using Exhaust for Heating—We are operating our plant with a 750-kw. turbine, 150 lb. boiler pressure and 100 deg. superheat. Over one-half of our total requirements for steam go for heating and drying of paper, and we reduce live steam to 1-lb. pressure for our heating and drying purposes. We have a 300-hp. steam engine standing idle. Could we not get better economy by starting this engine up, relieving the turbine of 300 hp. and using the exhaust steam for heating and drying purposes? B. W.

It will be more economical to supply your heating and drying apparatus with exhaust steam from the 300-hp. reciprocating engine than to have that power developed by the turbine and supply the heating and drying apparatus with live heat. In thus passing the steam through the reciprocating engine it will part with only enough heat for performing the work and heat lost by radiation, together with such as may be lost in discharged drip water. On the whole you should, by proper appliances for drainage, etc., be enabled to realize 75 to 80 per cent. of the heat thus passed through the reciprocating engine, in addition to obtaining the power developed, besides relieving the turbine of the load carried by the reciprocating engine.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

A Study of Boiler Losses*

By A. P. Kratz†

SYNOPSIS—Results of numerous tests made to permit of a detailed study of the boiler and furnace losses under varying conditions of load, depth of fuel bed, and draft.

The boiler on which these tests were run is one of a battery of two Babcock & Wilcox boilers. The battery, as a whole, has 10,160 sq.ft. of heating surface, equally distributed between the two boilers, and is designed to carry a working pressure of 160 lb. per sq.in. Each boiler has two 42-in. by 20.33-ft. drums and 18 sections of 4-in. tubes 18 ft. long, each section containing 14 tubes. They are set so that the distance from the floor to the first row of tubes is 8½ ft. at the front end.

A cross-section through the boiler is shown in Fig. 1. It may be seen that a style of baffling different from the standard Babcock & Wilcox type is in use. This construction pro-

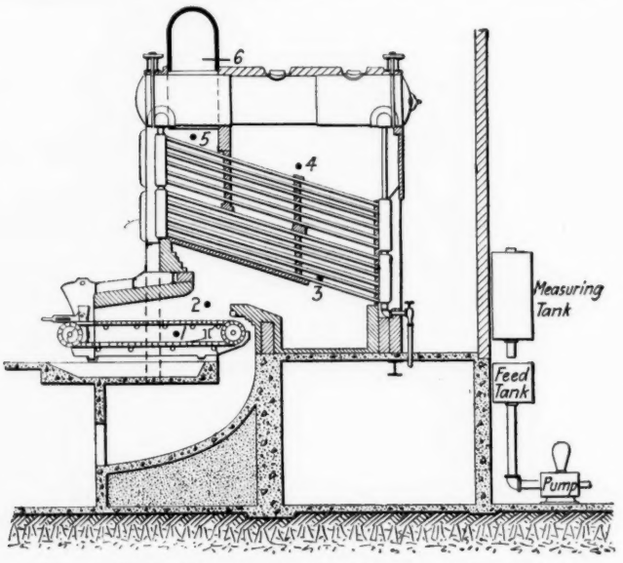


FIG. 1. SECTION THROUGH BOILER AND FURNACE

vides a tile roof over the furnace and makes the attainment of a long combustion chamber possible.

The first pass for theg ases is thus at the rear of the setting, and they are discharged into a breeching at the front instead of at the back of the boiler, as in the standard type.

The stokers are of the chain-grate type, having an active grate surface of 90 sq.ft., and were built by the Green Engineering Co. At the time these tests were made, each stoker had a flat fire-arch 6 ft. long, set 10 in. above the grates at the front end and 18½ in. at the back. This arch has since been changed so that at present it is 15 in. above the grates at the front and 33 in. at the back.

The draft is produced by means of a brick chimney 175 ft. high, having an internal diameter of 10 ft.

Flue-gas samples were collected over half-hour periods. One sample was taken at the breeching about 2 ft. below the dampers and a second one at a point about 8 in. behind and below the end of the tile roof of the combustion chamber. This point was about 34 in. back of the bridge-wall at the point where the gases turned to go up through the first pass. It was close enough to the furnace so that no air leakage chargeable to the boiler setting had as yet occurred, and far enough back so that combustion was complete, as may be seen from a comparison of the CO in the gas there and at the dampers.

Curves are shown in Figs. 2 and 3 in which efficiency is plotted against thickness of fuel bed. The best efficiency under full load conditions was obtained with a fire of about 7 or 7½ in. in thickness. This seems to give a fuel-bed resistance such that the normal amount of coal can be burned without excessive draft. If a thinner fire is used there is a greater probability that there will be holes or thin places in the fuel bed, forming paths of least resistance, through which an excessive amount of air passes with a consequent decrease in efficiency. On the other hand, if the thickness is materially increased it becomes necessary to use more draft to maintain the load. The increased draft causes an increase in the air leakage through the setting, etc., thus causing the efficiency curve to drop. The foregoing discussion is based on the assumption that there is no more CO formed when using a thick fuel bed than with a thin one. An inspection of the gas analyses shows that the CO was

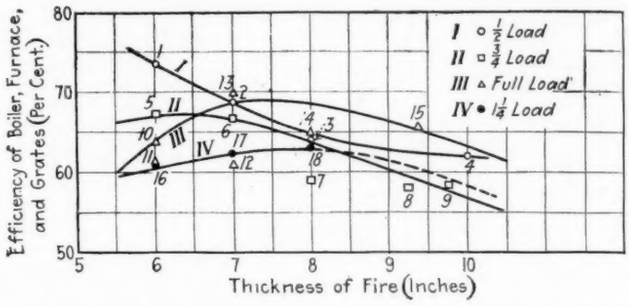


FIG. 2. RELATION BETWEEN THICKNESS OF FIRE AND EFFICIENCY OF BOILER, FURNACE AND GRATE

never more than 0.24 per cent. as a maximum and that this value was obtained on but two tests. For this reason it has been disregarded.

Another explanation of the decrease in efficiency when the thickness of fire is increased above 8 in. is offered by the fact that the arch is very low, being only 10 in. above the grate at the front end. This causes an excessive draft loss toward the front and makes the combustion poor over that part of the grate, thus reducing the effective grate surface. On this account it was found impossible to obtain loads as high as 1¼ capacity with a fire 9½ in. in thickness. That part of the 1¼ load curve to the right of the point representing the 8-in. fire has therefore been shown as a dotted line and has been made to parallel the full-load curve.

A study of the curves in Figs. 2 and 3 will show that for each load there was a well-marked thickness of fire that

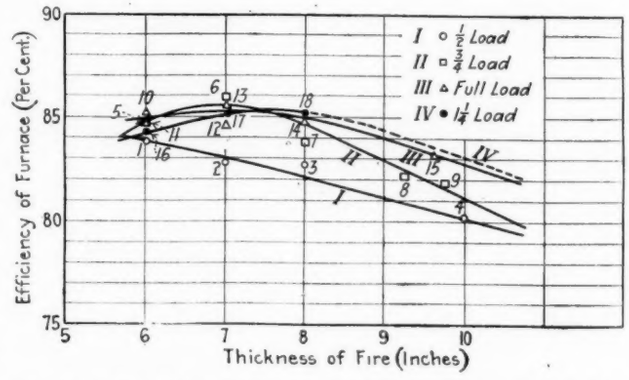


FIG. 3. RELATION BETWEEN THICKNESS OF FIRE AND EFFICIENCY OF FURNACE

gave the best efficiency and that as the load decreased this also became less. Thus for 1¼ load the maximum efficiency occurred with 8- to 8½-in. fires, for full load with 7- to 7½-in. and for ¾ load with about 6½-in.

The curves in Figs. 4 and 5 show the relation between efficiencies and capacity or load in boiler horsepower. The efficiency of the furnace and grate was fairly constant for all loads and depths of fuel bed. The loss in efficiency at higher loads was due to reduced heat absorption by the boiler. In general, the lower boiler efficiencies correspond with the higher temperatures.

*Abstract of Bulletin 78 of the Engineering Experiment Station, University of Illinois.

†Assistant in mechanical engineering, University of Illinois.

The curves in Fig. 6 give the drafts throughout the boiler setting. The positions plotted as abscissas refer to the draft-gage positions indicated in Fig. 1. It may be noticed that there is an unexpected drop in the curves at positions 4 and 6. At first it was supposed that this was due to error either in the gage itself or in the position of the tube. All the gages were then calibrated and found to be well within the limits of accuracy. The positions of the tubes were also changed, and they were moved backward and forward across the channels, with the same result each time. On comparing the cross-sectional areas of the various passes, it was found that

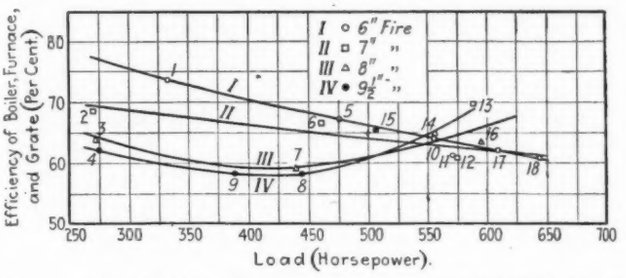


FIG. 4. RELATION BETWEEN LOAD AND EFFICIENCY OF BOILER, FURNACE AND GRATE

the area at point 4 was greater than that at point 3; also that the wall forming part of the last pass was corbeled so that the area at point 6 was greater than that at point 5. This peculiarity in the curves therefore has been attributed to the dynamic effect of the flowing gases.[1]

Referring to Fig. 7, it may be seen that the reading of a draft gage h'_1, when connected to a chamber in which the pressure is less than atmospheric, indicates the difference between the barometric pressure H and the true pressure or static head h_1, measured above absolute zero. If a Pitot tube is placed at the same point, and there is a flow of gas, the velocity head h_3 will add to the static head h_1, and the reading of the Pitot tube h'_2 will be less than that of the draft gage h'_1. According to Bernouilli's theorem, the total head at any point is the pressure head, plus the velocity head, plus the head lost up to that point. The atmospheric pressure H is the total head in this case, and h_2 is the sum of the pressure and velocity heads. Therefore h'_2 must represent the lost head or, in other words, the friction head. Proceeding from the front of the furnace to the breeching, the quantity h'_2 increases more or less uniformly, and since it is a loss, it would deduct from the value of h_1 and increase the gage

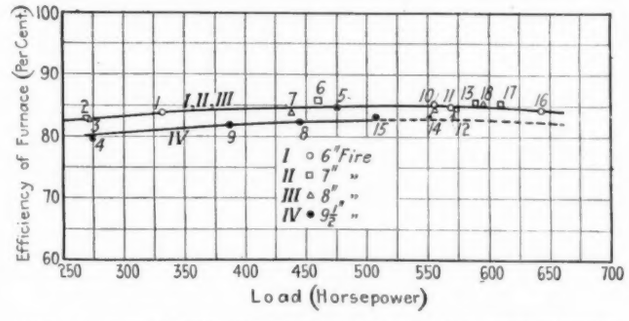


FIG. 5. RELATION BETWEEN LOAD AND EFFICIENCY OF FURNACE

reading h'_1. In view of this fact, the natural expectation would be to find the gage readings increasing for the successive positions from the furnace to the breeching. Any decrease in velocity head h_3, however, goes into an increase in pressure head h_1 and therefore makes the gage reading h'_1 less. Consequently, if a material increase in section or reduction of gas volume occurs at any point, the decreased velocity head might diminish h'_1 enough to make it read less than the gage preceding it. This explains the dip in the curves at positions 4 and 6 in Fig. 6, since at position 4 there was an increase in section over that at position 3, and also at position 6 over that at position 5. It may therefore be seen that the practice of subtracting draft-gage readings and calling the result the friction loss is incorrect unless the area of cross-section is the same at both points where the readings were taken. If this is not true, a correction for velocity should be made.

[1]For further discussion of this action in boilers see "Draft in Furnaces and Flues," by E. G. Bailey, "Power," Nov. 9, 1915.

Three factors determine the amount of draft necessary—the thickness of fire, the amount of dust in the coal and the horsepower developed. Taking each load separately, it was seen that the 9½-in. fire invariably required the maximum draft.

The relation between rate of combustion and the horsepower developed may be represented by a straight line.

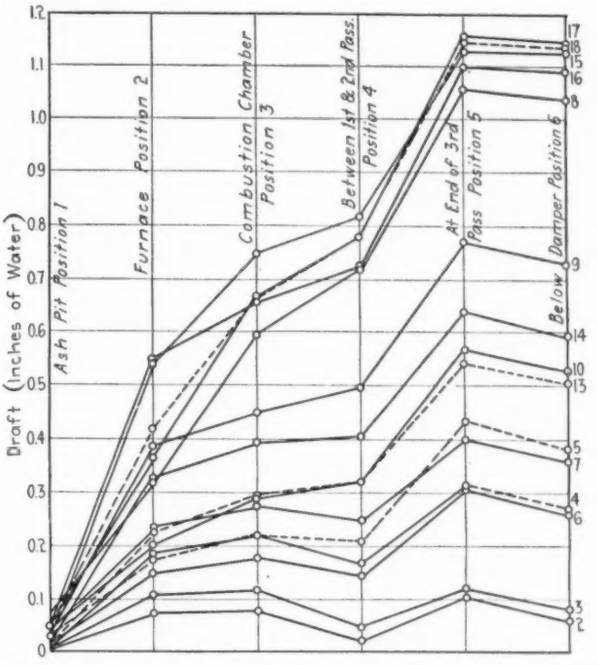

Draft (Inches of Water)
Ash Pit Position 1
Furnace Position 2
Combustion Chamber Position 3
Between 1st & 2nd Pass. Position 4
At End of 3rd Pass Position 5
Below Damper Position 6

FIG. 6. DRAFT PRESSURE THROUGHOUT BOILER SETTING

The curves in Fig. 8 give the relations between excess air, thickness of fire, and load. There are a number of other factors which affect the excess air, and the curves shown are at best only approximate. They are influenced by draft conditions, which in turn are somewhat dependent upon the dust content of the coal.

Taking the curve for each load separately, it is evident that the percentage of excess air decreases to a minimum and then

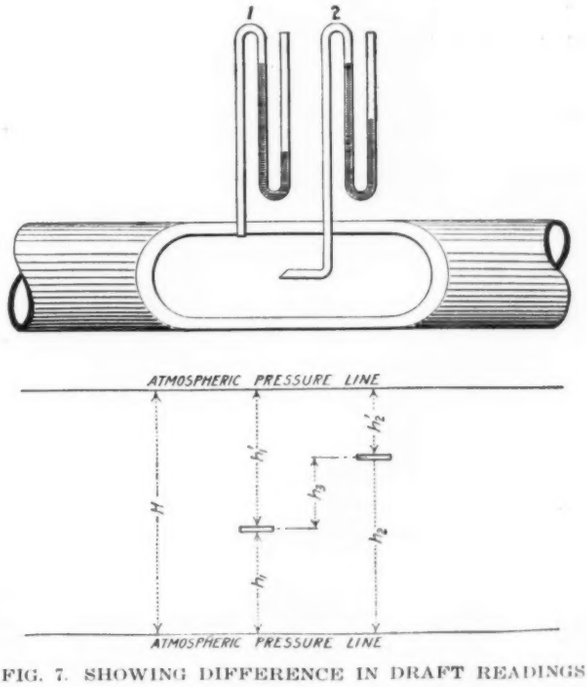


FIG. 7. SHOWING DIFFERENCE IN DRAFT READINGS

increases again as the thickness of fire increases. With thin fires there is a marked tendency toward the formation of holes in the fuel bed, through which an excessive amount of air can pass. Also, the entire fuel bed is more open and porous, and the small channels formed by the voids are

shorter. Hence the air is not brought into as intimate contact with the incandescent surface of the coal as it otherwise would be. When the thickness of the fire is increased, the passage becomes more tortuous and thus a larger proportion of the volume of air passing through the fuel bed comes into contact with the surface of the coal. At the same time the total area of the incandescent surfaces increases. Hence the excess air decreases. As the depth of fuel bed increases, however, the ash, as it forms, interferes more with the processes of combustion. At a thickness of about 7½ in. it appears that the ash begins to protect the incandescent surface from the action of the air, thus reducing the total area exposed and serving to mitigate the effect gained by the increased turbulence of the air. Therefore, increasing the depth of fuel on the grate beyond this point results in an increase in the percentage of excess air.

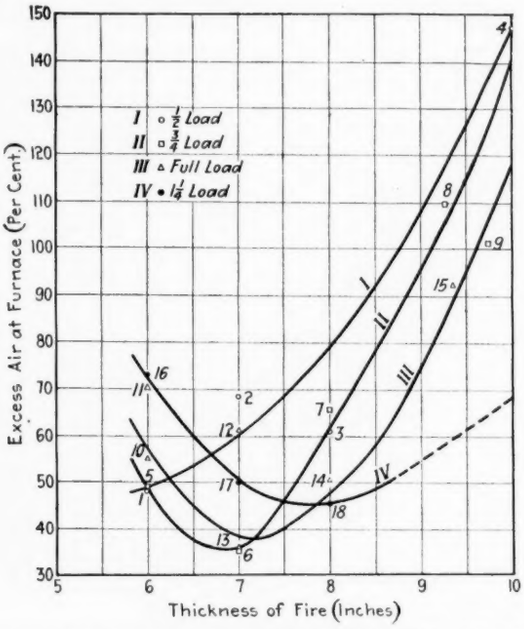


FIG. 8. RELATION BETWEEN THICKNESS OF FIRE AND EXCESS AIR IN THE FURNACE

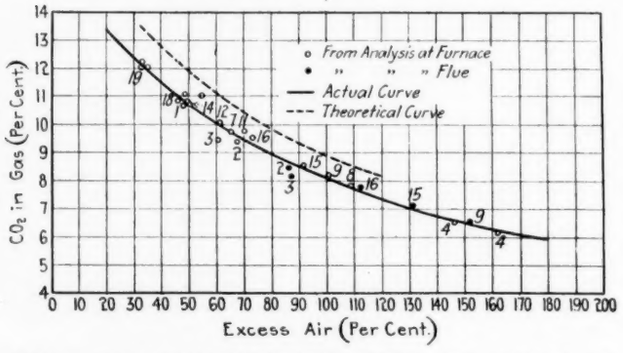


FIG. 9. RELATION BETWEEN EXCESS AIR AND CO_2 IN THE GASES

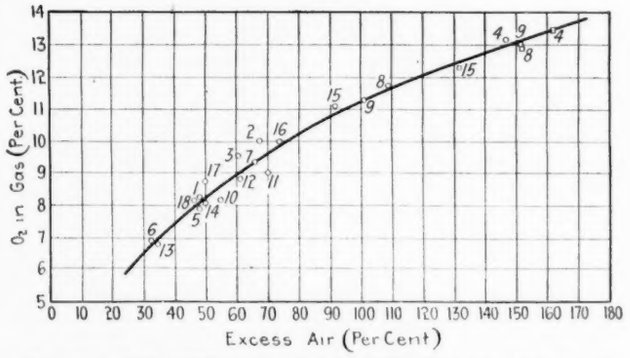


FIG. 10. RELATION BETWEEN EXCESS AIR AND O_2 IN THE GASES

Inspection of the curves in Fig. 8 will show that for fires less than 7½ in. thick, the excess air increases when the load or rate of combustion is increased if a constant depth of fuel bed is maintained, while for fires above 7½ in. thick the reverse is true. When the load is made greater, a larger volume of air is drawn through the fuel bed. That part actually coming into contact with the fuel also increases but, in the case of thin fires, not so rapidly as the total. Hence in this case the excess air will increase with load. As the fires become thicker, the part coming into contact is a larger proportion of the whole and it is not necessary to increase the total volume as much to obtain a given rate of combustion. At about 8 in. the tendency for the ash to choke the fuel bed becomes manifest, and with fires above this thickness, when the draft is made greater, the increased velocity of the air tends to sweep the surfaces clean and to permit better combustion. Hence the percentage of excess air is apt to be lower at the higher rates when using thick fires.

The trend of these curves is also influenced somewhat by air leakage along the sides and at the back of the stoker. This would have had a more marked effect when the heavier drafts were used. A study of the percentage of CO_2 at the furnace, however, from which the excess air was calculated will prove that this was at no time excessive. On all but two of the tests the CO_2 ran from 8 to 13 per cent., representing an excess of from 100 to 25 per cent., and in most cases was about 50 per cent.

In Fig. 9 the percentage of CO_2 in the flue gas has been plotted against the excess air expressed in per cent. Most of the points for this curve were taken from the analysis of the gas at the furnace, a few of the analyses at the flue being used, however, to establish some of the more remote points.

The theoretical curve in Fig. 9 was calculated from the following analysis, which was obtained by averaging the analyses for tests 1 to 18 inclusive:

C=65.77 per cent., H_2=4.17 per cent., O_2=6.27 per cent., N_2=1.65 per cent., S=4.52 per cent., and Ash=17.62 per cent. On the assumption that complete combustion took place, a gas analysis was computed for the coal mentioned using 100, 75, 50, and 25 per cent. excess air, the results of which are shown as a dotted line in Fig. 9.

The fact that the curve plotted from the analyses so closely parallels the theoretical curve may be taken as an indication of the accuracy of the gas analyses made on the tests. The explanation for the theoretical curve falling above the other lies in the fact that some of the carbon in the coal actually used went through the grates, some went to form soot and a small amount to form CO. This made the carbon actually appearing in the gas per pound of coal about 5 per cent. lower than that used for the theoretical curve.

CARBON DIOXIDE AND CARBON MONOXIDE RELATIONS

In Fig. 10 the O_2 in the flue gas is plotted against percentage of excess air. This gives the same results as Fig. 9, and of the two, the latter is the more valuable, since readings of CO_2 may be easily obtained in practice, while O_2 is not so readily determined.

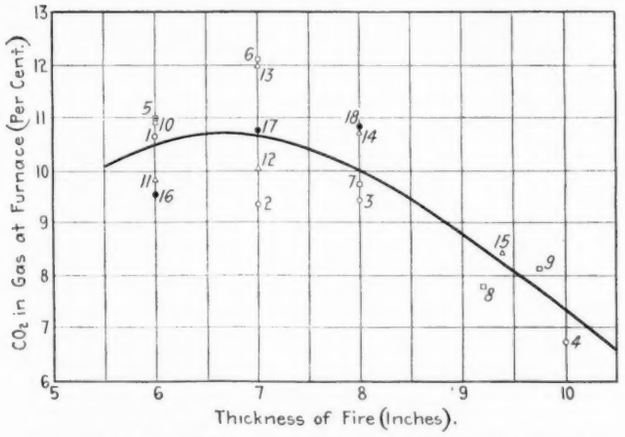


FIG. 11. RELATION BETWEEN THICKNESS OF FIRE AND CO_2 IN THE GASES

Fig. 11 was plotted in an attempt to obtain a relation between the thickness of the fire and the percentage of CO_2. The latter seems to have a tendency to reach a maximum on the 7-in. fire, from which point it falls off rapidly as the thickness of the fire is increased.

Tests indicate that the air leakage through the setting is roughly proportional to the draft over the fire. This leakage

is affected by the variation in the pounds of gas per pound of coal, which is in turn influenced by a number of factors previously discussed.

An attempt was made to find a relation between the loss due to excess air and the loss due to CO formed, but it was

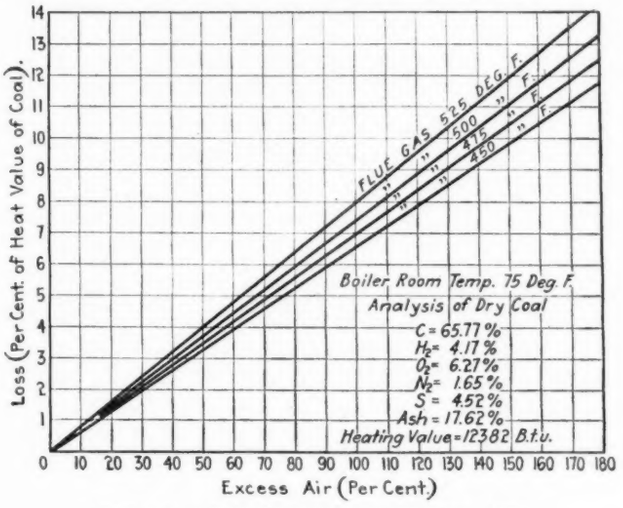

Loss (Per Cent. of Heat Value of Coal).
Excess Air (Per Cent.)
FLUE GAS 525 DEG. F.
" " 500 " F.
" " 475 " F.
" " 450 " F.
Boiler Room Temp. 75 Deg. F.
Analysis of Dry Coal
C = 65.77 %
H2 = 4.17 %
O2 = 6.27 %
N2 = 1.65 %
S = 4.52 %
Ash = 17.62 %
Heating Value = 12382 B.t.u.

FIG. 12. LOSS CAUSED BY EXCESS AIR

found impossible to deduce such a relation from the data obtained.

The curves in Figs. 12 and 13, however, have been calculated by making use of the coal analysis given under the discussion of Fig. 9. This analysis is fairly representative of the coals used. From these curves some idea may be gained of the relation between the losses caused by excess air and CO. Fig. 12 has been calculated for the different flue-gas temperatures indicated, on the assumption that the gases enter the furnace at 75 deg. F. and that the temperature of the flue gas would be the same for the different percentages of excess air.

Assume that under certain operating conditions 11.5 per cent. CO_2 is being obtained and that there is no trace of CO in the gas. Reference to Fig. 9 will show that 40 per cent. excess air is being used. Assume that the air supply is cut down, and when the percentage of CO_2 becomes 11.9, indicating 35 per cent. excess air, a trace of CO begins to appear. If the air supply is further reduced, so that the percentage of CO_2 becomes 12.9, indicating that only 25 per cent. excess air is being used, reference to Fig. 12 will show that the loss has been cut down from 2.61 per cent. when using 35 per cent. excess air, with the flue gas at 500 deg. F., to 1.88 per cent. when using 25 per cent. excess, with the flue gas at the same temperature, giving a difference of 0.73.

From Fig. 13 the percentage of CO which represents a loss equal to 0.73 is 0.19 when 25 per cent. excess air is being used. Hence, in the case assumed, if 0.19 or more of CO appears in the gas when the excess air is reduced from 35 per cent. to 25 per cent., the gain from reducing the air supply is more than offset by the loss due to the formation of CO.

For every furnace and for every coal, therefore, there will be well-defined limits beyond which the air supply cannot be reduced without causing a greater loss due to incomplete combustion. Just how much CO is permissible for any given coal and installation may be determined by plotting a set of curves similar to those given, and by finding by experiment at what value of CO_2 a trace of CO begins to appear.

The relation between loss due to combustible in ash and loss due to excess air is influenced by so many factors and the points fall so irregularly, that it is impossible to draw any very exact conclusions. The general tendency may be fairly well represented by the straight line. This shows that for the chain-grate stoker the tendency is for the loss due to excess air to increase as the combustible in the ash is decreased, and that the sum of the losses remains fairly constant at from 12 to 14 per cent. This means that to reduce the carbon appearing in the ash it is necessary to increase the air supply, which in turn entails a corresponding loss. It is probable that the combustible loss, being a visible one, will cause more concern than the invisible excess-air loss, but the foregoing in some measure indicates the futility of trying to cut down on the former at the expense of the latter.

Fig. 14 will give some idea as to the relative magnitude of the several losses. In this figure the heat balances for five typical tests have been plotted, from which it appears that the controllable losses due to the gases escaping at a temperature higher than that of the steam are in all cases less than the unavoidable losses up to the temperature of the steam. About the only method of diminishing these losses is by reducing the excess air and eliminating air leakage, since these are the only factors that are not at a minimum.

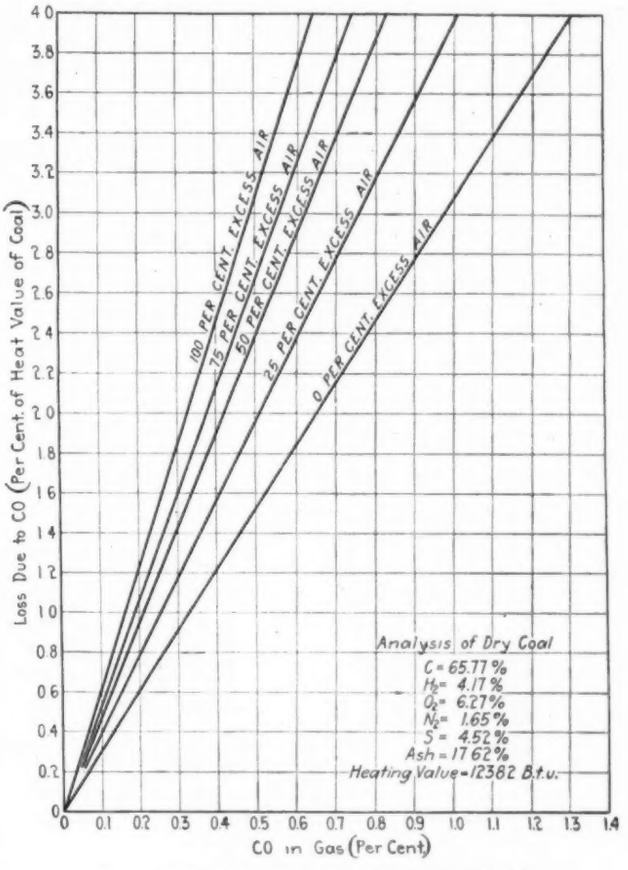

Loss Due to CO (Per Cent. of Heat Value of Coal)
CO in Gas (Per Cent)
100 PER CENT. EXCESS AIR
75 PER CENT. EXCESS AIR
50 PER CENT. EXCESS AIR
25 PER CENT. EXCESS AIR
0 PER CENT. EXCESS AIR
Analysis of Dry Coal
C = 65.77 %
H2 = 4.17 %
O2 = 6.27 %
N2 = 1.65 %
S = 4.52 %
Ash = 17.62 %
Heating Value = 12382 B.t.u.

FIG. 13. SHOWING LOSS CAUSED BY CO

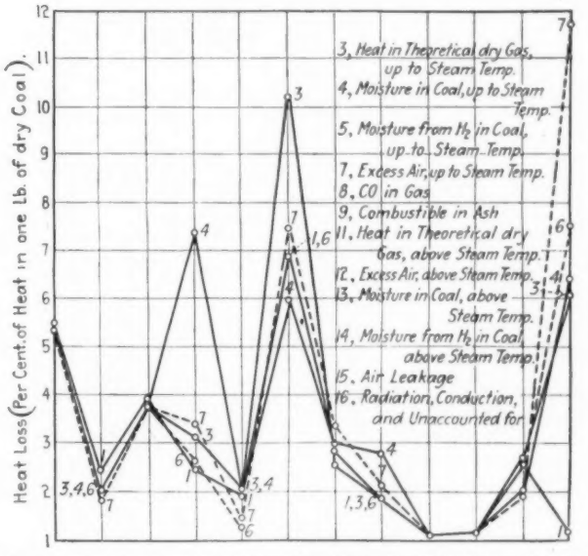

Heat Loss (Per Cent. of Heat in one Lb. of dry Coal).
3, Heat in Theoretical dry Gas, up to Steam Temp.
4, Moisture in Coal, up to Steam Temp.
5, Moisture from H2 in Coal, up to Steam Temp.
7, Excess Air, up to Steam Temp.
8, CO in Gas
9, Combustible in Ash
11, Heat in Theoretical dry Gas, above Steam Temp.
12, Excess Air, above Steam Temp.
13, Moisture in Coal, above Steam Temp.
14, Moisture from H2 in Coal, above Steam Temp.
15, Air Leakage
16, Radiation, Conduction, and Unaccounted for

FIG. 14. RELATIVE MAGNITUDE OF THE VARIOUS LOSSES

The largest loss seems to be due to the combustible in the ash. In a general way, however, where this is high, the excess-air loss is low.

Since the unaccounted-for loss includes the effects of various experimental errors, as well as radiation and conduction, it has no definite relation to the other losses. It can be reduced, however, by adequate heat insulation for the setting.

A study of the boiler efficiency and furnace efficiency shows that these two are very nearly alike, indicating that the losses are about equally distributed between them.

Elements of Specifications*

By Morgan B. Smith†

SYNOPSIS—An exact specification is the best safeguard of the rights of both purchaser and producer. The author points out the requisites of contract specifications and offers valuable suggestions on specification writing, with examples.

One of the greatest advances in the matter of purchasing materials and apparatus made in recent years has been the adoption by many companies of the so-called Specification and Test Basis for purchases. By the Specification and Test Basis we mean that the purchaser and the producer or seller agree, generally in writing, upon all the characteristics of the material or apparatus desired by the purchaser; and, secondly, they mutually agree to accept or reject the material or apparatus according to the results of tests previously agreed upon by both parties.

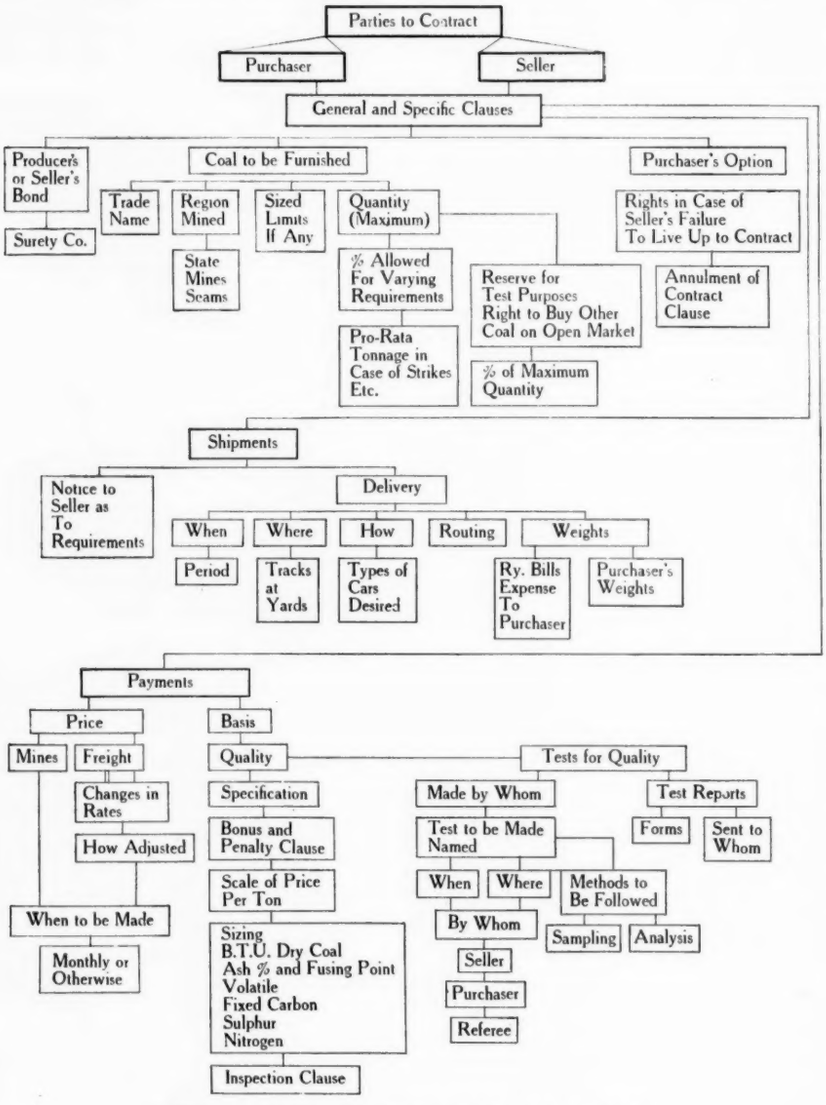


FIG. 1. OUTLINE SPECIFICATION OF COAL

The greatest benefit from this procedure has arisen from the fact that far less misunderstanding and misinterpretation of the terms of the agreement have been encountered since its adoption as the basis for purchase. Litigation has been avoided, and both the purchaser and the producer have benefited.

WHAT A SPECIFICATION CONSISTS OF

A specification consists of a set of precise provisions, each of which defines and fixes the character of one element desired. It must define (a) the utility or work to be done and (b) the business end of the subject matter. The former (a) supplements and explains the original plans or the so-called tentative letter as to the character of materials and methods and often the main principles and rules drawn in accordance with the utility and the market available. The latter (b) defines the rights and duties of each party to the specification, with due provisions for changes in the plans, for settlement of disputes, payments, acceptance clauses, etc.

You will at once notice that a specification by its very nature possesses all the essentials of a contract, and it is with this view that we must regard a specification. To go deeply into the matter of contracts is impossible within the time available. Suffice it to say that contracts are the heart of our business activity.

There was a time when specifications were looked upon as clubs to be wielded by the purchaser over the producer's head in the effort to obtain some commodity. Much sharp practice was the inevitable result. Fortunately, we have begun to see this matter in its true light and specifications are now regarded as agreements between the purchaser and the producer whereby both parties are afforded a clear, definite understanding of the requirements of the case at hand and disputes and misunderstandings are rendered less likely.

What are the essential factors of a specification contract?

First—Competency. Each party to the contract must be capable of fulfilling all the terms of the contract.

Second—Mutuality. The parties to the contract must have mutually agreed to all the terms of the proposed contract.

Third—Legality. The subject matter must be lawful.

Fourth—Valuable consideration. There must be the element of cost, which, according to the authority Johnson, is "some right, interest, profit or benefit accruing to one party or some forbearance, detriment, loss or responsibility given, suffered or undertaken by the other party."

We find two general classes of contract—the sealed and the parole contract. In either the valuable consideration must be proved, and in the long run the written contract is better, since its terms are reduced to writing and nonfulfillment of such a contract is less likely.

In any case fraud vitiates all contracts and consists in any misrepresentation by either party to the contract. Likewise a contract will not hold good if either party has been influenced by duress caused by threat of personal or property injury or deprivation. Both parties must assent to its terms of their own free will.

What are the essential points of view when writing a contract specification?

First, the legal; second, the technical; third, the rhetorical.

As a rule we do not find it particularly difficult to keep the legal and technical viewpoints in mind. But how do our specification contracts fare in the matter of rhetoric? Here is where we often fall down, for, strange as it may seem, the average engineer often is not adept in the use of good, clear and definite English. Many engineers therefore furnish the necessary technical data to their attorneys and let them handle the wording of the subject matter. The specification end must be handled by the engineer in so far as technical data

*Abstract of a paper read by the author before the Detroit Engineering Society, Sept. 17, 1915, entitled "The Purchase of Materials and Apparatus on the Specification and Test Basis."

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are concerned; the legal end by the attorney; the rhetorical end by both, working in harmony.

The rhetorical treatment must be such that neither misunderstanding nor any misrepresentation or misinterpretation of terms can become possible. What, then, are the characteristics of good rhetoric?

First—The meaning intended. There must be no room for doubt.

Second—The writing must be complete, avoiding unfinished sentences and including all essentials without superfluous matter.

Third—Accuracy. Definiteness in details, bearing in mind that it is the specific rather than the general, the exact and not the vague which we are after.

Fourth—Conciseness, but never at the expense of clearness or of completeness.

How, then, shall we proceed to write such a specification contract?

There is no better method than the so-called comparative, in which examples of written specifications are collected, compared and sifted for the best material for your given problem. The steps taken are, in order: Collection of material, classification of data, comparison and organization of material, and, finally, reconstruction and improvement. The engineer is thus afforded the benefit of the experiences of others, and, most important of all, he will place himself in a self-criticizing attitude toward his own compositions.

We must admit, however, that it is sometimes difficult to follow the comparative method because copies of existing contracts are not available. This may arise from the matter of personal rights, of publicity and often because the other fellow is somewhat over-modest with regard to the value of his own documents. In such a case the engineer is left to his own resources.

He must get as broad a view as possible of the requirements and should confine himself to but one phase of his subject at a time and thresh it out as thoroughly as possible before taking up another. He should bear in mind that good specification contracts are the result of long thinking and often of costly experience. He should not be discouraged if his first draft is a failure when tried out in practice, for

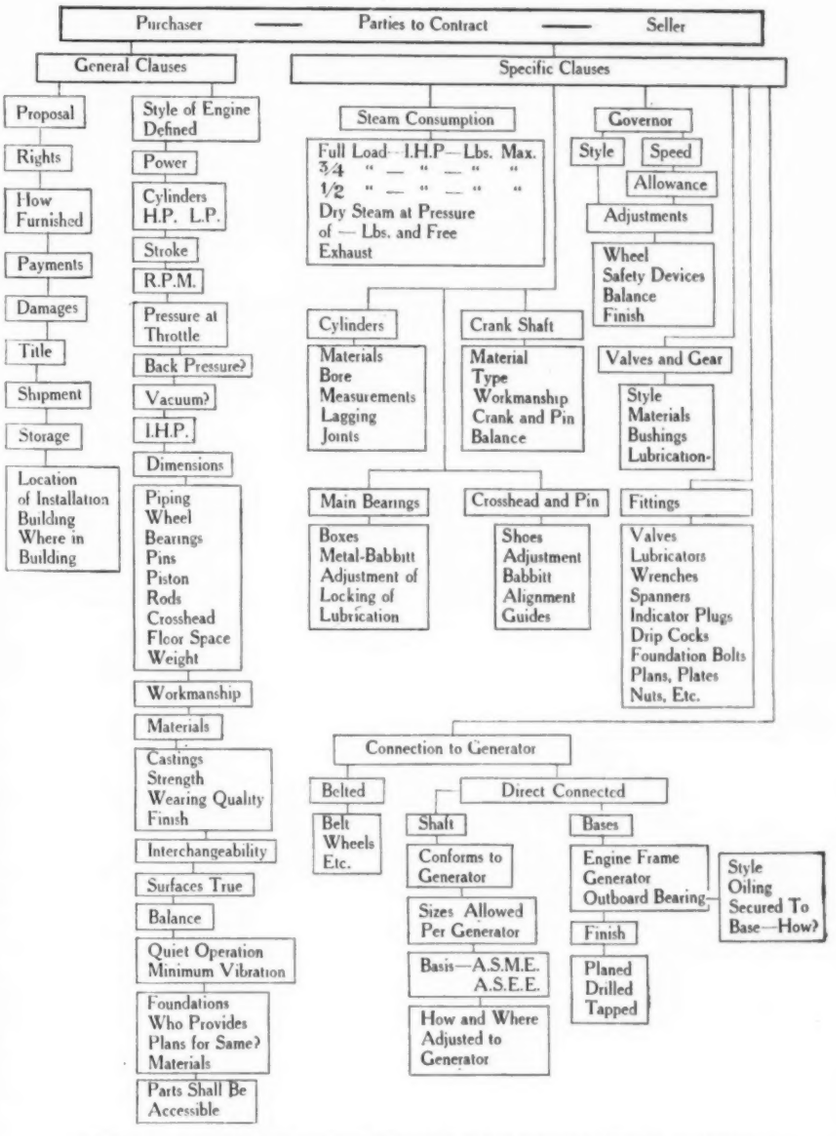

FIG. 3. OUTLINE SPECIFICATION OF RECIPROCATING ENGINE

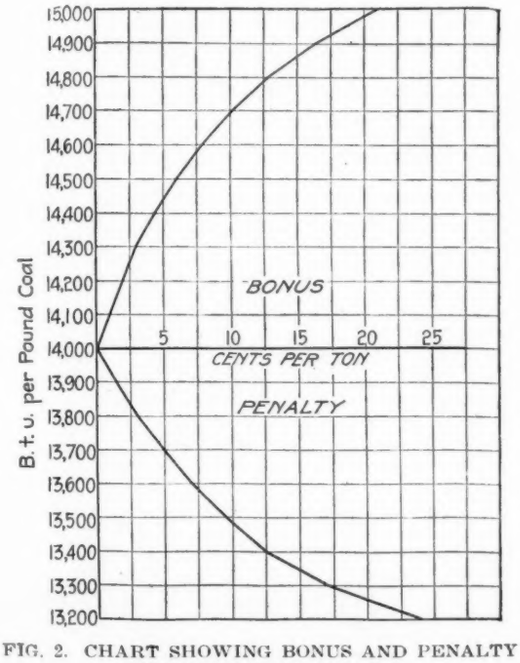

FIG. 2. CHART SHOWING BONUS AND PENALTY FOR QUALITY OF COAL

good drafts are the result of a process of elimination, generally involving considerable cost of time and money. The author should submit his efforts to a soaking process whereby his contract becomes cooked through. There is no more inefficient contract, nor one more likely to produce trouble, than a half-baked specification.

Beware of inherited specifications in their entirety; look them over carefully; there may be a nigger in the woodpile to be chased out. Improvements in existing specification contracts have come as the result of loss suffered through hazy or incomplete subject matter.

Before taking up a few examples of specification contract writing, it is well to emphasize again the necessity of both parties having a clear and definite understanding of the work to be done and a good knowledge of the market materials or apparatus available.

If possible the purchaser should adopt standard materials or apparatus available in the market, otherwise he generally invites mistakes that are costly.

In the long run it is the wise fellow who assumes that the manufacturer knows his own business a little better than one outside of that particular line of work. The manufacturer's judgment is invaluable to the purchaser and should be sought. To get at the opinion and judgment of the manufacturer the purchaser has recourse to the so-called tentative letter, which embodies a fairly definite proposal of the matter in hand, giving such data as will enable the manufacturer to pass judgment upon the merits of the case. Such a letter must indicate that you, the purchaser, know what you are writing about, for then and then only will your letter possess a certain valuable psychological value in its effect on the manufacturer.

Competition is desirable. Be "open" to the market as a rule. If you desire to "sew up" your specification, do it in a straightforward manner, and be sure that your treatment of the subject is ethical.

Should you find that there is no standard article on the market to fit the conditions of your utility, get in touch with the market before drawing up your specification. In a great many cases you will find the manufacturer able to meet your requirements by comparatively simple adaptations of articles already standardized, a procedure generally very inexpensive when compared with the working out of your original conception of an article entirely outside of standard practice. This, we must admit, cannot always be done. As a matter of fact, we must agree that some of our greatest advances have come through conceptions foreign to standard practice—the very essence of invention. Invention spells industrial progress.

Bear in mind the truth that so-called best practice is largely mutual practice between purchaser and producer in the matter of the original purchase and in its subsequent service. In other words, your specification contract to be enduring must be more than skin deep.

Let us now consider a few examples of specification contract writing.

OUTLINE SPECIFICATION CONTRACT

One of the best methods to follow in writing such a draft is to first outline the principal elements to compose your finished contract. With such a form it is easier to note omissions and to make needed additions than it is when many sheets of writing are before you. I propose therefore to show you my conception of the outline specification contract by means of the following illustrations. These may not meet all conditions in any case, but will at least serve to illustrate the principle involved.

One of our most widely used materials, coal, has received considerable attention, especially by the Federal Government, which has met with some success in its purchases of this commodity. This success is more difficultly attained by private purchasers than by the Government, because private interests are not and cannot be in the position occupied by our Government relative to producers. The principal obstacle seems to be the penalty and premium clauses usually incorporated in such contracts. These clauses are generally very distasteful to the producer, who feels that he ought not to be penalized for his shortcomings and who is rather skeptical in regard to the matter of premiums when he does a little better than required. Or putting it plainly, he feels that he may not be duly credited when premiums are justly due him. Experience has proved, however, that where the producer has been convinced of the sincerity of the purchaser, and that it is to his best interests to satisfy the purchaser, a mutual understanding and endeavor can be attained. In such cases there is seldom need of penalty and premium clauses in the contract.

For power-plant purposes it is generally found best to specify the heating value per pound, the ash content, the sulphur and the fusing point of the ash. The last is a measure of the safe working temperature of the fires in your furnaces. For some utilities, as in coal for byproduct coking, it is often necessary to specify the nitrogen content as well. Unless your plant is furnished with the necessary coal-preparing machinery, such as crushers, you will also have to specify the sizing of coal furnished you. If you are not equipped with track hoppers, you should specify flat-bottomed cars because of the difficulty found in handling bottom-dump cars.

In Fig. 1 is a chart for guidance in drawing coal specifications; Fig. 2 is a type of chart for showing penalties or premiums for different heat values of coal.

We could enumerate many examples of materials suitable for specification. Let us, however, consider apparatus. But can we altogether leave the subject of materials when thinking of apparatus? Hardly, for all of our apparatus is, after all, but the proper assembly of materials to form the finished machine. The familiar reciprocating engine forms a splendid example of this truth, and in Fig. 3 a typical specification contract is outlined.

[The author gave illustrations of similar charts for guidance in drawing specifications for cements, electric wire, steel rails, car axles and various other supplies.]

SPECIFYING A VENTILATING FAN

Probably no more hazy specifications have been written than those supposedly covering fans for ventilating purposes.

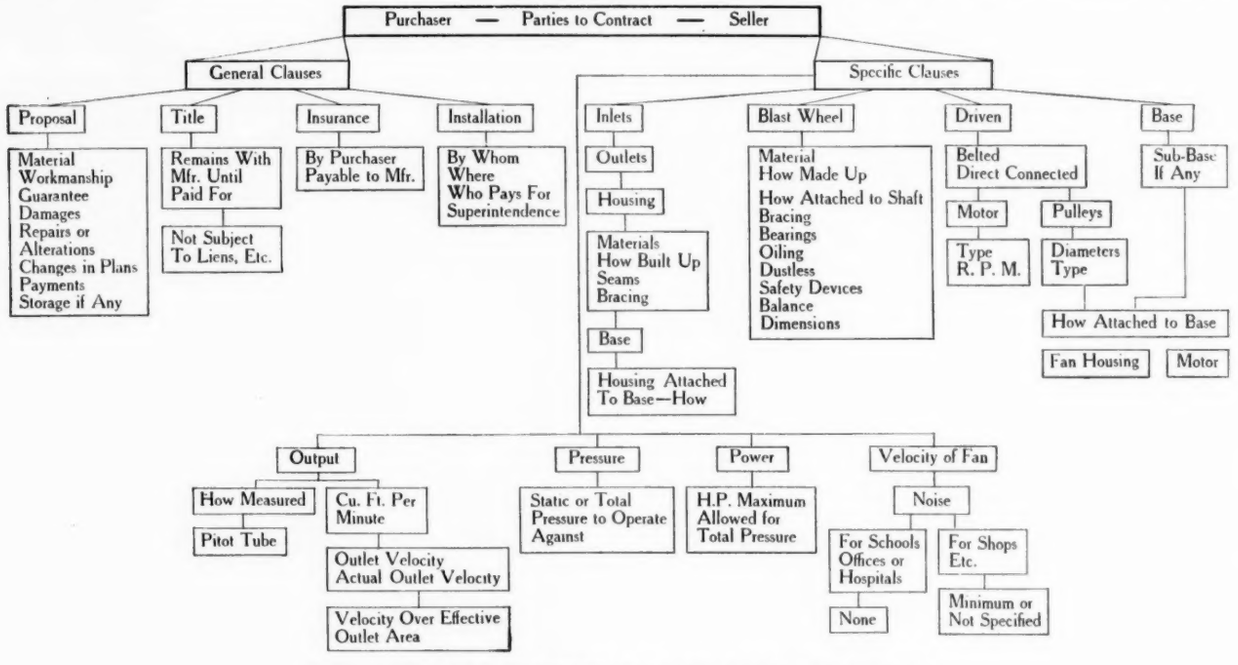

FIG. 4. OUTLINE SPECIFICATION OF VENTILATING FAN

An example is found in the omission of data in regard to the speed of the motor for driving the fan. Noise depends largely upon the speed of the fan. As a rule, where noise must be minimized high speed is out of the question. A moderate or low speed is essential, which in turn involves the design of the fan itself if the output is to be up to specification. If you do not want noise in operation, you must say so in your specification. If you must have a certain output under all conditions, you should say so. You must also give the manufacturer head against which the fan must deliver its output at the stated speed, etc.

Fig. 4 is an outline of such a specification contract.

SPECIFICATIONS OF TESTS OF MATERIALS

In specifying tests of materials or apparatus, both parties should agree upon methods of testing, for the use of unlike methods has caused much trouble between producers and purchasers. In testing coal difficulties arise over the preparation of samples and in the matter of test methods and types of laboratory apparatus used. It is not absolutely necessary that the same sort of apparatus, for instance, calorimeters, be used, provided dissimilar types have been calibrated against one another and their comparative results are known for similar work.

In the reports of tests, methods, however, should always be stated, otherwise, in many cases, results are almost meaningless to one not acquainted with the procedure in your laboratory. Laboratory tests must, as far as possible, duplicate practice conditions if we may hope to get real value from the results.

SPECIFICATIONS OF TESTS OF APPARATUS

The same reasoning as for materials here holds good, namely, that so-called acceptance tests made under conditions other than those that the apparatus must meet in service are generally worthless. Far better to hook up your machine to its intended load and then by your tests determine its exact behavior toward that load.

For example, what is its action under a constant load? What under a varying load? What under suddenly increased loads taxing it to the utmost? Will it hang on until you can put other machines on the work, or will it lie down at once and leave you "high and dry"? Factory tests have their value, but the tests made after installation in your factory are of the greatest value to the purchaser.

While much that I have said is known to those of you who are engaged in this line of work, I believe the outline form for specification contract writing has not been so generally used as its merits warrant.

To summarize, purchases for materials and apparatus can be made profitable for both purchaser and producer, provided a mutual understanding exists as to requirements and tests.

Increase in Safety of Boiler Operation in Prussia

The author attempts to prove that the number of boiler accidents in Prussia has decreased during the last few years.

He divides all accidents into three groups—explosions, accidents that lead to the boiler being put out of operation, and various defects producing accidents. In the main, disturbances in operation are produced by one of the following four causes: First, defects in material and workmanship; second, piping, connections and auxiliary apparatus; third, attendance; fourth, operating conditions. He reports his data in the form of tables and curves, some of which are here reproduced. Fig. 1 shows the number of boiler explosions caused by imperfections in material and workmanship, referred, for purpose of comparison in various years, to a unit of 10,000 boilers. The great jump in the number of explosions between 1894 and 1897 is due to the fact that during that time a different definition of boiler explosion was used by the statistical office.

Under the class of explosions due to boiler piping, connections and auxiliary apparatus are all accidents due to jammed or overloaded safety valves, clogged or incorrect manometers,

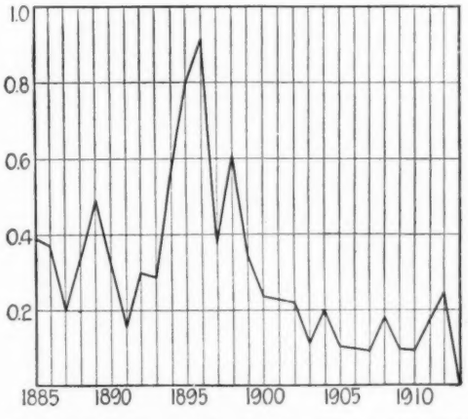

FIG. 1. EXPLOSIONS (PER UNIT OF 10,000 BOILERS) CAUSED BY DEFECTS IN MATERIAL AND WORKMANSHIP

defective feed-water apparatus; also accidents due to the clogging of passages leading to the water gages and incorrect indication of water level by the gages. The curve shown indicates that the number of accidents reported as due to this cause is comparatively small and varies widely in different years, which, under these conditions, may be due to accidental causes.

The number of accidents due to lack of proper attendance appears to be comparatively large. Explosions on these cases are often due to lack of water in the boiler, which can be traced to the carelessness of the fireman. Fig. 2 indicates the gradual falling off in the number of explosions. The author points out a curious phenomenon, namely, that statistics indicate that the number of accidents due to careless attendance on boilers has increased in the last few years, while the

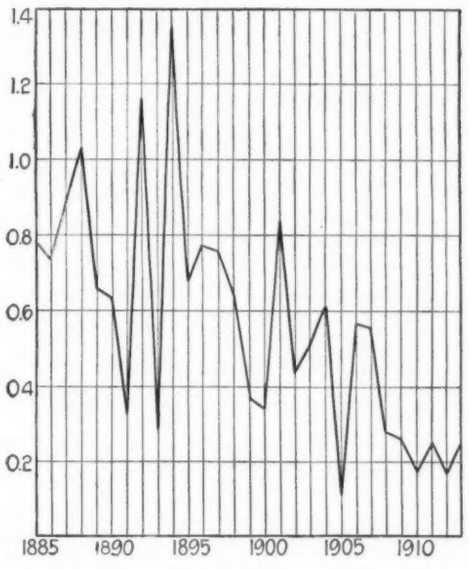

FIG. 2. SHOWING THE DECLINE IN THE NUMBER OF EXPLOSIONS

number of explosions due to this cause has decreased. Since the majority of accidents due to this cause are produced by low water, it would appear that the boilers are being so built that they can better withstand trouble of this nature. The table offers an opportunity of judging the tendencies in explosions and accidents in boiler plants by giving a comparison between what took place from 1885 and 1889 on one hand, and from 1909 to 1913 on the other. The same data are rep-

COMPARATIVE DATA SHOWING VARIATION IN NUMBER OF BOILER ACCIDENTS IN PRUSSIA IN LAST 18 YR.

Causes	Explosions to Each 10,000 Boilers in the Period 1885-9	1909-13	Accidents to Each 10,000 Boilers in the Period 1885-9	1909-13
Defects in material and workmanship	0.352	0.120	4.48	1.78
Defects in piping connections and auxiliary apparatus	0.246	0.069	2.04	1.84
Defective attendance	0.815	0.225	5.83	6.82
Defective operation	0.351	0.135	5.26	3.90

resented by curves in the original article. ("Untersuchungen über die Zunahme der Sicherheit der Dampfkesselbetriebe in Preussen," B. Hilliger, Zeits. des vereines deutscher Ingenieure, Vol. 59, No. 34, p. 681, Aug. 21, 1915, 7 pp., 10 figs.).—"The Journal of the American Society of Mechanical Engineers."

Friction of Lubricated Surfaces approximates to that of "solid friction" as the bearing becomes dry and to that of "fluid friction" as it is flooded with oil.

One Hundred Year Old Engine—According to the Glasgow "Herald," a letter from the Farme Colliery was read at a meeting of the Corporation of Glasgow, stating that they had in their colliery at Rutherglen an old "Atmospheric" engine, which was to be superseded by one of modern construction. It was an interesting and unique relic of the early days of steam, and would no doubt be of interest to the city with its engineering industry. The firm offered it to the Corporation on the understanding that it would be placed in one of the parks. It was agreed to accept the offer and thank the donor. The "Atmospheric," or Newcomen, engine has been at work at the Farme Colliery drawing coal since it was erected in 1809 until a month ago. During that time, with the exception of one or two spur-wheels that were broken by accident, no part has been renewed. Until it was superseded by a unit of modern construction, a month ago, it was the oldest engine at work in Scotland, and the only "Atmospheric" engine at work in Great Britain.—"Engineering," London.

New Orleans Secures Reduction in Electric Rates

As a climax to a two years' fight between the city officials and the New Orleans Railway and Light Co., a new schedule of rates has been offered by the company and accepted by the city council to become effective Dec. 1. Throughout the entire negotiations the city stood firm and threatened to build a municipal plant if its contentions were not met. The new schedule is as follows:

Commercial and Residential Lighting Rates: First 20 kw.-hr. in any one month, 7c. per kw.-hr.; next 30 kw.-hr. in any one month 6c. per kw.-hr.; next 150 kw.-hr. in any one month, 5c. per kw.-hr.; all in excess of this, 4c. per kw.-hr. There is an additional service charge of 25c. for each meter. Minimum bill to be 50c. per month.

Retail Power: Primary, 6c. per kw.-hr.; secondary, 3½c. per kw.-hr.

The rates are net, to which the company may add 10%, which amount it shall offer as a cash discount.

Besides the reduction to private customers a material reduction will be made in the cost of city lighting, and the electric company agrees to pay for the construction of an entirely new street-lighting system. It is estimated that the new rates will at first make a difference of about $300,000 in the yearly receipts of the company, but it is believed that much of this will later be offset by increased business.

Boiler Capacity per Family

In a lecture before the Detroit Engineering Society, J. C. McCabe, city boiler inspector, gave the following interesting data as a prelude to a plea for an engineering building in Detroit:

Statistics indicate that there is in use one steam boiler for power purposes in any community in the civilized world for every 225 to 250 of population. There are 2,700 steam boilers in use in Detroit, aggregating 425,000 hp. and averaging individually over 150 hp. In addition there are listed about 250 stationary boilers installed, but not in use and about 400 portable boilers that may or may not be again used in this city. Consequently there is one boiler in use for about each 250 population, giving 0.6 hp. to each person, and if the family unit is taken at five persons, there will be 3 hp. to each family. Inasmuch as the power of a man to work is taken at ⅛ hp., it will be seen that the use of steam makes available to each family in Detroit the equivalent of the work of 18 men.

In the United States today there is steam apparatus doing work that would require the efforts of about 350,000,000 men to perform. This does not include farm engines, railroad locomotives or steamboats. The ambition of the engineer is to improve the condition of man by making available for his use an increasing multiple of his own ability to perform useful labor.

Detroit is a city of boundless opportunity and will become increasingly prosperous and great as the multiple of work done mechanically to the unit of population is increased by engineering means. The men that make it possible to draw together the engineers, chemists and architects of the city for the closer interchange of opinion and experience will be the benefactors of the race. Why should not Detroit have a distinctive engineering building, housing all organizations of engineers, chemists and architects and others connected with engineering matters?

First Meeting of Chicago A. S. M. E.

On Nov. 19 at the Hotel La Salle the Chicago section of the American Society of Mechanical Engineers held its first meeting of the season. It was a dinner session, as usual, and about 125 were in attendance.

Frank B. Gilbreth, an authority on scientific management, delivered an illustrated lecture on "Motion Study." The speaker told how the movements of the worker were photographed and studied with a view to lessening the effort expended and increasing the output. His introductory remarks touched upon the work he is doing for the warring countries in Europe. Mr. Gilbreth is conducting a systematic campaign for the instruction of teachers in this new form of vocational guidance and vocational training, for the placing of men and, in particular, the enormous number of cripples from the war.

Hon. Jacob M. Dickinson, former Secretary of War, followed with a few remarks on the weaknesses of the pension system of this country and related how some of the cripples of the Civil War took up their old or new occupations, some of them making better records than their previous accomplishments would have indicated.

Naval Men Witness a Test of Refrigerating Machine

Many prominent naval and shipbuilding officials were present at East Pittsburgh recently to witness a test of the Le Blanc refrigerating machine manufactured by the Westinghouse Machine Co.

A feature of the machine is the absence of all chemicals or compressed gases such as carbon dioxide, ammonia or sulphur dioxide. The presence of tanks of carbon dioxide under pressure is an element of danger on battleships when likely to be struck by an enemy's projectile. With this machine the cooling of the water or liquid is accomplished in the machine itself without the use of a cooler.

ENGINEERING AFFAIRS

Steam-Boiler Code—Thomas E. Durban, chairman of the American Uniform Boiler Law Society, says that the adoption of the code for the design, construction and operation of steam boilers, evolved by a committee of the American Society of Mechanical Engineers, has made progress to the following extent: Boilers built under this code will be acceptable to the boiler departments of the states of Ohio and Indiana, the City of Detroit and the City of Chicago, and the authorities in the states of Pennsylvania, Wisconsin and California have accepted the code, to be put into force later. The society of which Mr. Durban is chairman is in correspondence with a number of other states that are preparing laws dealing with steam boilers. The legislatures of Georgia, Kentucky, Louisiana, Massachusetts, New York, Rhode Island, South Carolina, Virginia and New Jersey meet next year, and the efforts of the committee will be directed toward the adoption by these legislatures of uniform boiler-inspection laws based upon the American Society of Mechanical Engineers' Code.

The Combined Associations of the N. A. S. E., of Manhattan, Bronx and Queens held their fourteenth Annual Entertainment and Reception at Terrace Garden, New York, on Nov. 6. There were in attendance Walter C. Damon, national president; Fred Raven, national secretary; Samuel B. Forse, national treasurer; George W. Brownhill, Henry Mistele and William J. Bedard, national trustees; J. William Peterson, permanent secretary of the National Exhibitors' Association, and several of the national past presidents. In numbers the event was the largest in the history of the combination. The hall was neatly decorated. A vaudeville performance, including the New York Bunch of Entertainers, preceded the dancing. At 6 o'clock in the evening in the dining room of the Garden a complimentary banquet was tendered to the national officers and guests. The combination includes Brotherhood, No. 1; James Watts, No. 7; Progressive, No. 15; Protective, No. 23, Phœnix, No. 24; Manhattan, No. 25; German American, No. 29; Empire, No. 32; Stevenson, No. 44; Mott Haven, No. 47; Insurance, No. 56. The General Committee comprised George E. Whittemore, John H. Coors, William M. Logan and Arthur Jeynes.

Chicago Chapter American Association of Engineers—By-laws for the Chicago Chapter of the American Association of Engineers, with a membership of 160, were adopted at a meeting held Nov. 5. Fourteen standing committees are provided for: By-laws, meetings, program, entertainment, initiation, finance, inspection trips, publicity, articles for publication, membership, visiting, grievances, local legislation and employment and information. Chapters are being formed at Springfield, Ill., Minneapolis, Indianapolis and several other points. A provision is to be made in the constitution that after 500 members have been obtained an initiation fee of $2 will be charged, this to increase with each 500 new members. Arrangements will be made for a junior membership, which is to be extended to any applicant not eligible to certified membership because of qualification limitation. Such members will be entitled to all the benefits of certified members except the holding of office, the right to vote and being registered as engineers. When junior members have acquired the necessary qualifications, they may be transferred to certified membership. Junior members need not pay initiation fees upon joining nor upon transference to certified membership.